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## **Exhaust Gas Recirculation in Dual-Fuel Engine**

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**Abstract**

The efforts made during this thesis were pointed towards analyze and comprehend the effect promoted by exhaust gas recirculation systems within dual-fuel operation engines. The main outcomes of the study embrace the positive impact found in dual-fuel engines brought by low EGR rates regarding both thermal efficiency and exhaust emissions, as well as the effect promoted by changes of the methane share with respect the total amount of energy supplied.

The increase up to 50% of the diesel substitution rate leads to greater thermal characteristics, achieving higher cylinder pressure and cumulative heat release. Nevertheless, further increases of the methane share may produce significant reductions of the energy released associated to a poor combustion quality, as well as increase the likelihood of the onset of knocking.

The exhaust gas recirculation is mainly characterized by reductions of the oxygen content present within the intake mass flow. It is demonstrated how EGR rates below 15% bring notorious reductions of NO<sub>x</sub> without significantly compromising the thermal efficiency. Notwithstanding, when present to a large degree, the EGR may lead to erratic engine operation and, taken to the extreme, misfiring.

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**Keywords** Dual-Fuel, EGR, NO<sub>x</sub>, exhaust gas recirculation, substitution rate, combustion, dilution effect, misfiring, knocking.

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# Preface

This master thesis was carried out at the Department of Mechanical Engineering of Aalto University, under the supervision of Professor Martti Larmi as well as the guidance of Dr. Ossi Kaario and Dr. Teemu Sarjovaara.

I would like to express my gratitude to Professor Martti Larmi who, along with Teemu Sarjovaara, provided me the opportunity to conduct this research within the internal combustion engine research group. My sincere appreciation also goes for Dr. Ossi Kaario for his supervision and guidance along this period.

I am grateful to my colleagues at Aalto's combustion engine laboratory for their support during the whole term and their advices, which helped me to a large extent to develop the present study. They deserve many thanks for welcoming and accepting me as one of them as well as for making the lab a comfortable place to work in. I would like to bestow particular thanks to Otto Blomstedt and Olli Ranta for their help that in many situations guided me to accomplish my outcomes. My heartfelt appreciation goes for Rasmus Pettinen as well, whose cooperation and technical hints assisted me throughout the development of this thesis.

Thanks to my family for their constant support during my career and the confidence instilled in me ever since my early stages, specially to my parents Luis and Elena, as well as my brother Daniel for teaching me the values that made me the person I became over the years.

Espoo, 11 May 2016

Alejandro Calvo Oliveira

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# List of Abbreviations & Symbols

a: Crank Radius (m)

AFR: Air-Fuel Ratio

ATDC: After Top Dead Center ( $^{\circ}$ CA)

B: Cylinder Bore (m)

BTDC: Before Top Dead Center ( $^{\circ}$ CA)

BSEC: Break Specific Energy Consumption (MJ/Kwh)

BMEP: Break Mean Effective Pressure (bar)

CAD: Crank Angle Degree ( $^{\circ}$ )

CA: Crank Angle ( $^{\circ}$ CA)

Ch: Chemical Effect

$CH_4$ : Methane

CI: Compression Ignition

CNG: Compressed Natural Gas

CO: Carbon Monoxide

$CO_2$ : Carbon Dioxide

DF: Dual-Fuel

ECU: Engine Control Unit

EGR: Exhaust Gas Recirculation

EHVA: Electro-Hydraulic Valve Actuator

HC: Hydrocarbons

HRR: Heat Release Rate (KJ/°CA)

$H_2O$ : Water

l: Connection Rod Length (m)

LNG: Liquefied Natural Gas

$\dot{m}_{diesel}$ : Diesel Mass Flow (Kg/h)

$\dot{m}_{methane}$ : Methane Mass Flow (Kg/h)

$NO_x$ : Nitrogen Oxides

$O_2$ : Oxygen

PM: Particulate Matter

ppm: Parts-per-Million

PRR: Pressure Rise Rate (bar/°CA)

$Q_n$ : Net Heat Release (J/°CA)

Ra: Radical Effect

RME: Rapeesed Methyl Ester

SI: Spark Ignition

SOI: Start of Injection (°CA BTDC)

$SO_2$ : Sulphur Dioxide

SR: Substitution Rate (%)

Th: Thermal Effect

UHC: Unburned Hydrocarbons

VC: Combustion Chamber Volume ( $m^3$ )

VVT: Variable Valve Timing

$\lambda$ : Lambda/ Air-Fuel Equivalence Ratio

$\gamma$ : Isoentropic Constant

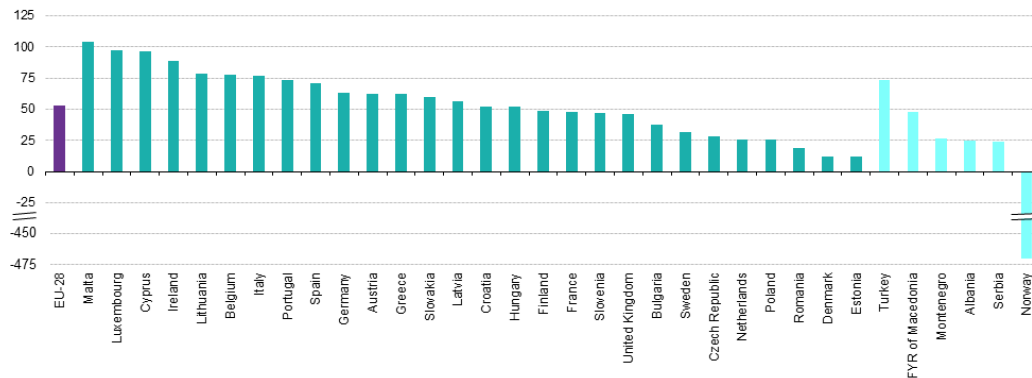
# 1. INTRODUCTION

## 1.1. Motivation

The continuous depletion of fossil fuels over the years along with the increasingly anti-environmental effects caused by pollutant emissions have been warning humankind against over-exploiting crude oil earth sources. Varying from one source to other, recent forecast last crude oil reserves nowadays explored for the next 40-50 years [21], which is promoting a constant increase in the price of petroleum. European countries are unable to hold the energetic demand with their own production sources. This issue relies on the fact that more than half of the gross domestic consumption in Europe (53.2% in 2013) [7] is provided by importation, as shown in figure 1.1.

Within the total energy importation, 13.97% stands for oil crude, which represented an investment of approximately 54.93 billion dollars in 2013 [3].

*Figure 1.1: Energy Dependency Rate, 2013*



Source: *producciones e importaciones de energía, eurostat, 2015,*  
<http://ec.europa.eu/eurostat/statistics-explained/index.php>

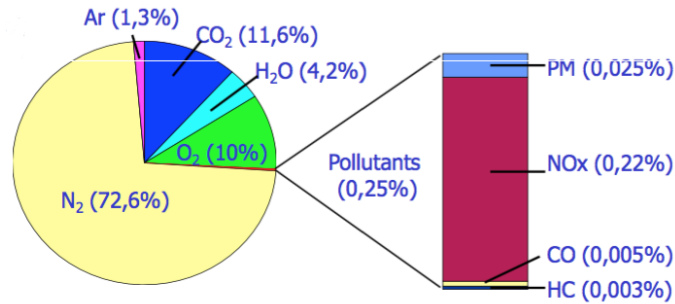
Aside from the energetic point of view, oil consumption leads to different gas emissions produced as a result of the combustion process. Emissions derived from combustion may be classified depending on the actuation zone into local or global



emissions. Currently, legislation and certification of vehicles take only local emissions into account, and therefore these receive the most attention from engine developers.

Global emissions, the main one being  $CO_2$ , contribute in a negative way to greenhouse effect, which entails to an acceleration of climate change. Local emissions nomenclature stands for HC, CO and  $NO_x$  emissions, being HC and  $NO_x$  the main problem when analyzing CI and SI engine emissions [14].

*Figure 1.2: Exhaust gases composition in CI engine*



Source: Estrategias para la reducción de emisiones, Motores Térmicos para Automoción notes, UPV, 2014

The motivation behind developing dual-fuel engines is the reduction of oil consumption by replacing gasoline/diesel fuels with natural gas, along with the increasingly tightening legislation forcing to reduce pollutant emissions. The use of natural gas in diesel engines has both economical and environmental advantages. The economic benefit stems from the availability of natural gas in huge quantities in many parts of the world, which used locally, might save the use of other type of fossil fuels (diesel/gasoline). Furthermore, natural gas gives high resistance to knock when used as a fuel in internal combustion engines. It is, therefore, suitable for engines of high compression ratio with allowing a possible improvement of thermal efficiency [26]. The environmental advantage stands for reduced particulate matter in the exhaust, as it contains less dissolved impurities than petroleum fuels, and the below carbon-to-hydrogen ratio of the gas is associated with lower emissions of carbon dioxide [19].

Either way, although dual-fuel technology is still on the first stages of maturity, is undoubtedly gaining greater importance within the internal combustion engine interests, thus several researches are being nowadays carried out in order to adapt legislation requirements to the upcoming times.

## 1.2. Objectives

The objective of this thesis is marked by the interest of analyzing the effect of the exhaust gas recirculation into the dual-fuel concept regarding both thermal efficiency and generated exhaust emissions. The operating points are decided to be varying basically regarding three parameters; % load, diesel substitution rate and EGR rate. As the EGR rate greatly influences the air-fuel equivalence ratio ( $\lambda$ ), it was carefully manipulated to actually run on the desirable range of  $\lambda = (1,2.2)$ .

As already mentioned, one of the interest of this research is to identify the released emissions to the environment and the understanding of their variations while working in a specified conditions. Due to the notorious effect of the EGR while treating pollutants, mainly  $NO_x$ , special attention is paid on this scheme, where the EGR system is re-designed attending to different needs along the present study.

For the execution of this master's thesis, and in order to get reliable data to analyze, several tests have been run in a dual-fuel engine with external EGR characterized for operating at high pressure levels (the stream is recirculated to the intake after the compressor stage). Dual-fuel engines typically are run with natural gas formed generally above 85% by methane and a small pilot amount of diesel that, due to its high reactivity, is used to initiate the combustion and thus, ignite the methane. During this study, methane content of the natural gas used is around 99%. Therefore it is foreseeable to accomplish higher energy output if compared with regular natural gas.

## 2. BACKGROUND

### 2.1. Natural Gas

During the eighteenth and nineteenth centuries, the industrial revolution impulsed to a large extent consumption of fossil fuels. Along that period, coal was the most popular source since its availability and energy value promised great advantages in terms of technological evolution. Coal exploitation lasted until mid twentieth century, when right after the second world war, was suppressed by oil consumption, which still nowadays is used to produce the major part of the global energetic consumption. In the three past decades natural gas has slowly but progressively increased its share of the energy mix. As Michael J. Economides and David A. Wood stated in 2009 on the article "The state of natural gas", these three fossil fuels account for more 85% of the world's primary energy [6]. The prospects for natural gas are so promising that the International Energy Agency (IEA) has suggested the 21th century could be the "Golden Age of Gas" with demand for natural gas projected to increase for over 25 percent of the world's energy supply mix by 2035 [25]. The table 2.1 reflects the evolution of total world energy supply from the last four decades.

*Table 2.1: Total World Energy Supply (Mtoe).*

Fuel Type	1973	2011
Natural Gas	16.0%	21.3%
Oil	46.0%	31.5%
Coal/peat	24.6%	28.8%
Biofuels and waste	10.6%	10.0%
Other sources	2.8%	8.4%

*Adapted from U.S. Energy Information Administration, U.S. Department of Energy, Washington, DC, 2014, <http://www.eia.doe.gov/emeu/international/reserves.html>*

Along with the increased demand for natural gas, comes a corresponding increase in international trade, which deals with the problem of transporting natural gas all over the world. To ease the shipment, natural gas is cooled to condensation at temperatures

of approximately  $-161^{\circ}\text{C}$  in order to get liquefied natural gas or LNG. This way, the energy per volume ratio is increased around 600 times, as the volumetric ratio between LNG and natural gas is about  $1/580$ , i.e. from the regasification of  $1\text{m}^3$  of LNG,  $580\text{m}^3$  of natural gas are obtained [8].

Average measurements indicate that one standard cubic feet of natural gas combusted releases 700 to 1600 Btu of heat depending upon gas composition [9], which varies according to the source from what the gas is obtained. The high methane level within natural gas composition provides great characteristics to the gas, such as high resistance to autoignition and knock [13]. A typical natural gas composition is shown in table 2.2.

*Table 2.2: Typical component and content of natural gas*

Component	Typical analysis (vol.%)	Range (vol.%)
Methane	94.9	87.0-96.0
Ethane	2.5	1.8-5.1
Propane	0.2	0.1-1.5
Isobutane	0.03	0.01-0.3
n-Butane	0.03	0.01-0.03
Nitrogen	1.6	1.3-5.6
Carbon Dioxide	0.7	0.1-1.0
Others	0.04	0.01-0.44

*Adapted from: A review on natural gas/diesel dual fuel combustion, emissions and performance, Lijiang Wei, PengGeng, 2015.*

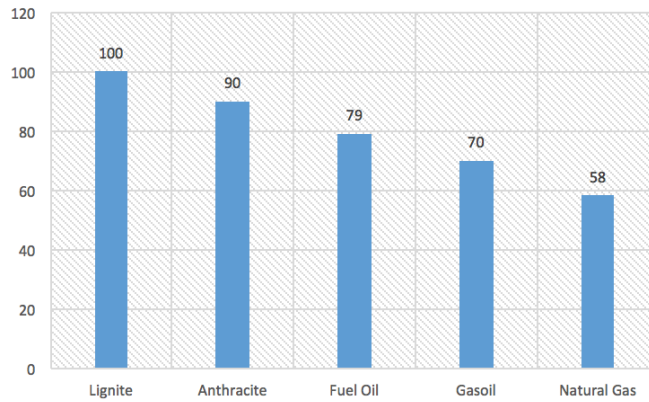
Within its benefits, natural gas produces the lowest environmental impact among all the fossil fuel. Important reductions of  $\text{CO}_2$ ,  $\text{NO}_x$  and  $\text{SO}_2$  emissions have been appreciated, as well as the absence of soot or any other kind of particulate matter when analyzing its emissions.

Natural gas, as the rest of fossil fuels, produces  $\text{CO}_2$  during the combustion process. However, due to its high H/C molecular percentage,  $\text{CO}_2$  emissions present a reduction around 40-50% compared to coal and 25-30% to fuel oil emissions [5], as can be seen in figure 2.1. The reduction of  $\text{CO}_2$  levels is beneficial in terms of reducing the greenhouse impact.

The composition of natural gas reduces the  $\text{NO}_x$  content into the exhaust gases to a large extent, reaching reduction values of 50% compared to coal and 40% to fuel oil emissions [5]. Hence, the utilization of natural gas also leads to a decrease into the environmental effects such as acid rain and photochemical smog.

Regarding  $\text{SO}_2$  emissions, sulphur content in natural gas remains below 10 ppm,

*Figure 2.1: CO<sub>2</sub> emission release for natural gas compared with different fossil fuels*



*Adapted from Industry, Energy and Tourism, ministry of Spain, Madrid, 2016,  
<http://www.minetur.gob.es/energia/gas/Gas/Paginas/gasnatural.aspx>*

this stands for reductions up to 150 times compared to gas-oil, between 70 and 1500 times for coal and about 2500 times lower than fuel-oil production [5]. As NO<sub>x</sub>, SO<sub>2</sub> represents an important causative of the acid rain and, as already mentioned, this phenomena may be greatly reduced by using natural gas as a primary fossil fuel source.

## 2.2. Dual-Fuel Engines

In 1901 Rudolf Diesel obtained a patent covering the concept of the dual-fuel engine. This modern type of engines, which are still nowadays utilized, are based on the use of a mixture of fuel and air compressed to a temperature below its autoignition temperature with the addition of an injection of a second, more reactive fuel with lower ignition temperature which promotes the ignition of the mixture [13].

The dual-fuel engine, not only consumes a wide range of gaseous resources effectively, but also has the potential to avoid much of the current and future problems facing the diesel operation, including the need for very significant reductions in its exhaust emissions [12] as well as the rising of the world energy demand [30].

The progressively wide availability of bulk supplies of natural gas made the use of dual-fuel engines fed with natural gas increasingly possible and economically attractive. As mentioned, one of the major advantages within the dual-fuel field resides on its fuel flexibility. They can be designed to operate interchangeably with primary fuels other than diesel using diesel as a pilot fuel, or they can be operated solely with diesel fuel. This flexible fueling strategy allows using renewable fuels easier, as conventional diesel fuel can be used in areas where no supply for renewable primary fuels is feasible [29].

Dual-fuel engines, when operating primarily with gas fuel, may have some aspects of their performance equal or superior to those of the common diesel engine while

economically exploiting a wider variety of fuel resources. However, although these systems have been operating satisfactorily and economically, there is still room for reducing costs and enhancing their performance further, whether in terms of efficiency, power production, maximizing diesel fuel replacement, displaying a wider tolerance to changes in gaseous fuel consumption or ensuring additional improvements in exhaust gas emissions [13].

### 2.2.1. Dual-Fuel Operation

Dual-fuel engine operation is based on the interaction of two fuels containing different reactivity level. The satisfactory operation of dual-fuel engines depends on numerous operating and design variables that tend to be greater in number than those controlling the efficiency of conventional spark ignition or diesel engines [13].

Currently there exist two types of operation within dual-fuel operation; Gas-Diesel direct injection system, in which both gas and liquid fuel are directly injected to the combustion chamber and the combustion is produced by the presence of air incoming to the cylinder, and conventional dual-fuel operation, based, as SI engines, on the premix of gas fuel with air into the intake manifold followed by the injection of diesel fuel into the cylinder when the premix is already established in the combustion chamber.

Gas-Diesel direct injection system is characterized by a limitation of 20% in diesel volume injection. The gaseous fuel is injected in the cylinder at high pressure (275 bar) and plays the role of the primary fuel, while the liquid fuel is considered as the ignition source. The limitation in  $NO_x$  and PM reduction compared with conventional dual-fuel operation have made this operation principle less attractive and its use is not as extended as the conventional operation [10].

Regarding the conventional dual-fuel operation, diesel percentage can vary around 0.5% and 100%. As mentioned above, the gas fuel-air blend is mixed into the intake manifold and introduced into the cylinder by means of the intake valves. When the compression stroke is approaching the top dead center, the diesel injection takes place causing the burning due to its higher reactivity level [10].

The implementation of dual-fuel engines is normally original from diesel engine arrangements and the installation of a gas injection system and a externally-fitted ECU must be carried out into diesel engines in order to implement the dual-fuel operation [10]. Dual-fuel solutions are becoming more attractive compared to CI engines due to its ability of better withstanding shock and knock loadings at high loads with low heat loss and good co-generation capacities [13].

Nevertheless, dual-fuel engines are always employed with very lean gaseous fuel-air mixtures, which can lead to the production of significant amounts of unburned hydrocarbons and carbon monoxide emissions [13]. Besides, it is known the interaction between the liquid fuel spray and the pre-mixed bulk of gaseous fuel-air greatly influences

combustion parameters such as ignition delay, as well as it may lead to increased emission levels.

### 2.2.2. Combustion Characteristics

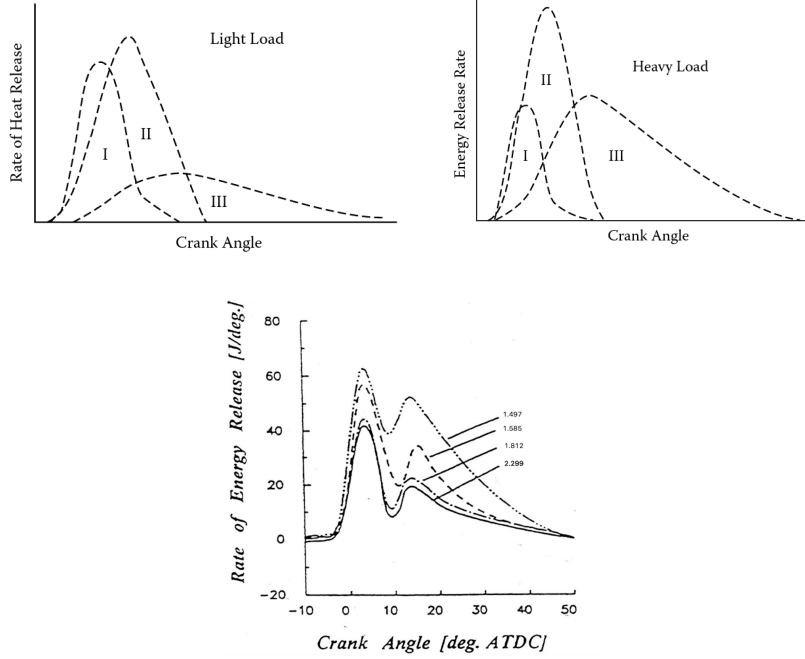
In dual-fuel engines, the primary fuel is premixed with air forming a homogeneous mixture (as in SI-engine) which is then ignited with a small amount of diesel injected into the cylinder at the end of compression stroke. As occurs in diesel engines, the ignition is produced by high pressures and temperatures by the compression. These similarities with CI and SI combustion processes made dual-fuel combustion broadly understood as a combination of mixture-controlled diffusion combustion known from CI engines, and turbulent flame propagation known from SI engines [29].

As G.A. Karim states in the article "Combustion in gas fueled compression ignition engines of the dual fuel type" [12], the dual-fuel combustion process depends both on the spray and ignition characteristics of the diesel pilot, the type of gaseous fuel used and its overall concentration in the cylinder charge. A limitation present during combustion regarding mixture conditions is that the turbulent flame propagation from the pilot ignition regions will not proceed throughout the charge until the concentration of the gaseous fuel is beyond a limiting concentration, in other words, within the very lean mixtures no consistent flame propagation will take place from the ignition centres and pilot influenced burning regions. In order to ease combustion in such conditions, an earlier pilot fuel injection may help in initiating combustion of lean mixtures since longer time becomes available for mixing the pilot fuel vapour with the gaseous fuel and subsequent ignition [13]. As the concentration of the gaseous fuel into the mixture is further increased, flame propagation throughout the rest of the charge resulting in a sudden increased contribution to the total energy release is achieved [12]. Nevertheless, a continued increase in the concentration of the gaseous fuel in air will result in increasing much of the energy release immediately following the commencement of the autoignition of the pilot. Such intense rapid energy releases may be associated with the onset of knock [13].

A general examination of the combustion process in a premixed dual-fuel engine can be made through the apparent heat release variation following G.A. Karim combustion analysis in his book "Dual-Fuel Diesel Engines" [13]. For convenience, the energy release rate may be considered to be made up of essentially three overlapping major components. The first, contributed by the combustion of the pilot. The second, due to the combustion of the gaseous fuel component that is considered to be in the immediate vicinity and influence the ignition and combustion centers of the pilot. The third, due to any preignition reaction activity and subsequent turbulent flame propagation within the overall lean mixture of mainly the gaseous fuel in air. Figure 2.2 shows a comparison between the energy release rate in both part and high loads considering combustion split in the three phases above mentioned, and the actual rate of heat release keeping constant the diesel injection quantity with variance of the excess-air

ratio.

*Figure 2.2: Heat Release Rate Comparison between G.A. Karim proposed Combustion model and real Combustion process*

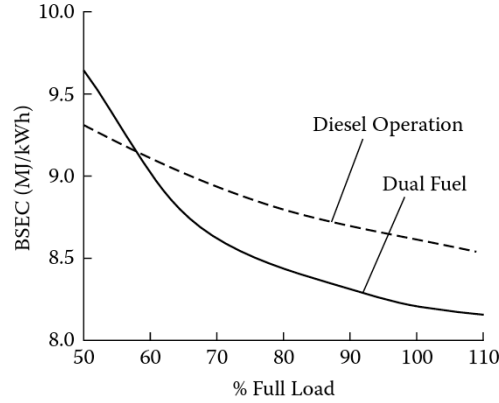


*Adapted from: Combustion in Gas Fueled Compression: Ignition Engines of the Dual Fuel Type, G.A. Karim, 2003*

Some limitations and challenges regarding load operation may be found in dual-fuel engines related to the combustion process. Concerning to high load operation, pre-ignitions and the onset of knocking attributed to a form of uncontrolled autoignition and subsequent very rapid combustion of part of the charge are the main facts limiting the dual-fuel operation range. With respect to part load operation, the fuel gas tends to be less completely oxidized at light loads, which tends to increase the specific energy consumption and exhaust emissions of those unburned hydrocarbons and carbon monoxide. However, these species that have been incompletely oxidized can play, through residual and exhaust gas recirculation, important chemical and thermal roles in the ignition and combustion of the subsequent cycles and may influence the associated extent of the cyclic variations. The kinetic effect of these residuals particles may result in increasing the activity of pre-ignition reactions during the latter stages of compression [13]. The figure 2.3 illustrates the variation of break specific energy consumption with the load for a dual-fuel engine operating on natural gas, and the corresponding values when operating as diesel.



*Figure 2.3: Variation of BSEC with % load for Diesel and Dual-Fuel engine operation*



Source: Danyluk, P.R., *ASME Journal of Engineering for Gas Turbines and Power*, 115, 728-733, 1993

Notwithstanding, other researches [4, 20, 22], express slightly different reasonings and results. For instance, R.G. Papagiannakis, Crookes RJ and Namasivayan AM agree in their studies that the thermal efficiency on dual-fuel engines closely resembles to the one experienced by diesel engines at high loads, coinciding in the superiority of the CI engine at low loads.

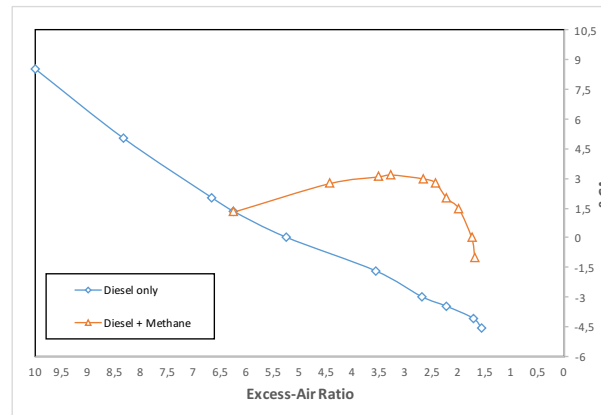
The disparity of resolutions conclude that stable and acceptable dual-fuel operation depends on the particular engine design and natural gas induction system used, as the original engine design parameters affect dual-fuel combustion significantly.

### 2.2.3. Ignition Delay

In dual-fuel engines, the ignition delay depends on many variables that need to be controlled to a large extent in order to optimize the combustion process and consequently the engine efficiency. This ignition delay in dual-fuel operation influences markedly combustion and is essential to keep it as short as possible in order to avoid reduced power output, low efficiency, increased emissions and undesirable high rates of pressure rise [13].

An ignition delay comparison between diesel and dual-fuel depending on the air-fuel equivalence ratio is shown in figure 2.4. It is appreciated how for dual-fuel operation, the delay tends to increase initially with the increased gaseous fuel admission up to a detectable maximum value, and then drops to a minimum well before reaching the total stoichiometric ratio based on the combined gaseous and liquid fuels and the available air [13]. However for diesel operation the delay decreases as the mixture concentration gets closer to the stoichiometric.

*Figure 2.4: Diesel and Dual-Fuel ignition delay vs Air-Fuel equivalence ratio*



*Adapted from: Dual-Fuel Engines, G.A. Karim, 2015*

The introduction of the gaseous fuel within the intake air produces variations in the physical and transport properties of the mixture, such as the specific heat ratio and to a lesser extent the heat transfer parameters. Besides, changes in the partial intake pressure of oxygen due to air displacement by the gaseous fuel, changes in the pre-ignition reaction activity and its associated energy release and the effects of residual gas can bring about substantial changes to the pre-ignition processes of the pilot and hence to the length of the delay period [12].

The EGR installation may affect the ignition delay as well by producing variations in the intake temperature and due to the dilution effect, which limits somehow the amount of oxygen available for combustion. These facts greatly influence the combustion temperature and consequently, the ignition delay.

Further techniques as advancing the injection timing of the pilot can generally lead, up to a point, to an earlier ignition [13].

#### 2.2.4. Knock in Dual-Fuel

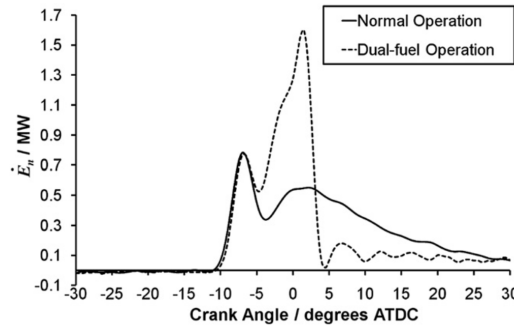
The energy release in dual-fuel engines normally involves the combined contribution of the pilot fuel regions together with the rapid turbulent flame fronts propagation originated from these regions. This mode of combustion that combines autoignition and rapid flame propagation is primarily responsible for the ability of dual-fuel engines to burn mixtures that are much leaner than those normally possible in gas-fuelled spark ignition engines. A prime requirement of any alternative gaseous fuel for satisfactory operation in dual-fuel engines is that its mixture with air would not auto-ignite spontaneously during or following the rapid pilot energy release. Failing to do so can lead to the onset of knock which manifests in excessively rapid rates of pressure rise and overheating of the walls leading to significant loss in efficiency with increased cyclic variations.

There is a tendency to encounter knocking with some gaseous fuels or under highly rated power output conditions [13] thus, knocking phenomenon restricts in some way the

load range in which the engine works properly. Persistent knock is highly objectionable and needs to be avoided; otherwise it may lead to engine failure [12].

As expressed by Crookes RJ [4], dual-fuel engines present shorter combustion duration (after pilot fuel ignition) than compression ignition engines, which stands for higher rise of pressure rates and an easily appearance of knocking. Figure 7 shows the comparison between DF and CI combustion duration by means the energy variation when rapeseed methyl ester (RME) is used as pilot fuel in dual-fuel operation with natural gas as main fuel.

*Figure 2.5: Comparison of net rate of energy change for dual-fuelling operation and normal engine operation.*



Source: *Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions*, Korakianitis, T. Namasivayam, A. M. Crookes, R. J., *Progress in Energy and Combustion Science*, 2011

The knock limited power output for any fuel and pilot setting has been shown to greatly deteriorate with the inverse of the intake absolute temperature [12]. It is also demonstrated that increasing the size of the pilot and variations of  $\lambda$  value influence the onset of knocking, although the effect of the last one tends to be relatively less severe [13].

Other factors such as the composition of the gaseous fuel mixture, the distribution of the gaseous fuel to air ratio by a proper stratification or the temperature of the cooling systems may help on the avoidance of knocking when properly controlled, however these techniques are off the scope of this study.

## 2.3. Exhaust Gas Recirculation

In internal combustion engines, exhaust gas recirculation is a technique utilized in order to decrease the  $NO_x$  emissions produced during the combustion process. By recirculating part of the exhaust gases, it is possible to achieve better combustion conditions, which have a direct effect into the exhaust emissions. Several mechanisms

have been developed along the years in order to attain faster and more effective responses. Nowadays, electronic EGR systems are installed to a large extent into engines mainly due to its high response speed.

The motivation behind the establishment of EGR systems appeared along with the increasingly concern about the effect of exhaust emissions into the environment.  $NO_x$  emissions affecting photochemical smog, incomplete combustion creating CO and HC that lead to adverse effects for human health, and impurities in the fuel and oil besides incomplete oxidation of hydrocarbons resulting in soot along with the appearance of  $CO_2$  as combustion product, are the motivating facts that give rise to a necessary pollutant control into the internal combustion engine field.

Different EGR strategies may be carried out according to the specifications of each case. The wide range of EGR strategies may be briefly classified following Molina Alcaide, S. criteria in his publication "Influencia de los parámetros de inyección y la recirculación de gases de escape sobre el proceso de combustión de un motor Diésel" [18]:

· *According the temperature level:*

Exhaust gas recirculation with cooling system is found in many cases due to the benefits of lower recirculation temperatures.

· *According the pressure level:*

The piping system might be installed into a high (behind turbine-after compressor) or low (after turbine-behind Compressor) pressure circuit.

· *According to the nature of the system:*

The setup may be either internal or external, being external configuration the most popular.

· *According to the mass of gas admitted:*

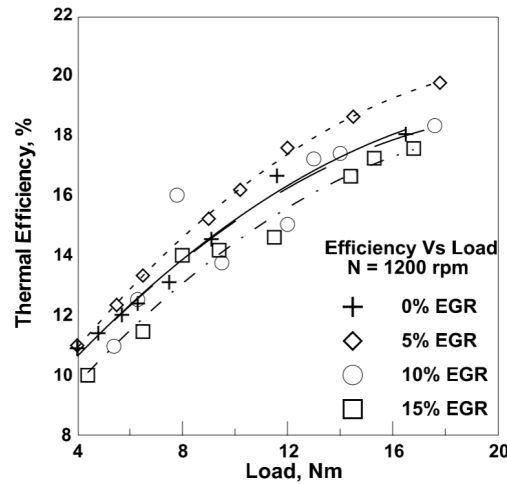
It is also frequent to find references of addition/substitution EGR, depending whether the intake mass of fresh air is kept constant or not.

The consequences of EGR installation into the engine may be explained regarding three main effects; chemical effect, thermal effect and dilution effect. Each of them affects in a different extent the combustion process and, hence, they have a direct influence on pollutant emissions.

The chemical effect is associated with the dissociation of  $CO_2$  to form free radicals and has been shown to be of minor effect. This fact may cause a slight increase in thermal efficiency attributed to the reburning of some hydrocarbons, especially at low EGR ratios [26].

The thermal effect corresponds with an enhancement of the combustion quality in terms of thermal efficiency by increasing the heat capacity of the mixture. The recirculation of part of the exhaust mass flow to the intake manifold promotes an increase of the inlet temperature. Since the maximum combustion temperature is closely related with the intake cylinder temperature, the presence of hot EGR leads to higher combustion temperatures and, hence, affects positively the thermal efficiency by increasing the combustion velocity arising from a higher intake charge temperature. However, this effect simultaneously generates an increase in NO<sub>x</sub> and particulate emissions compared with cold EGR since the combustion gases spend longer periods at these higher temperatures [26]. The figure 2.6 illustrates the effect of EGR into the thermal efficiency at constant speed. It can be appreciated how low EGR ratios affect positively the efficiency for all load ranges and the offset of this improvement by the dilution effect while increasing the EGR percentage.

*Figure 2.6: Thermal Efficiency variance with EGR and load at constant speed.*



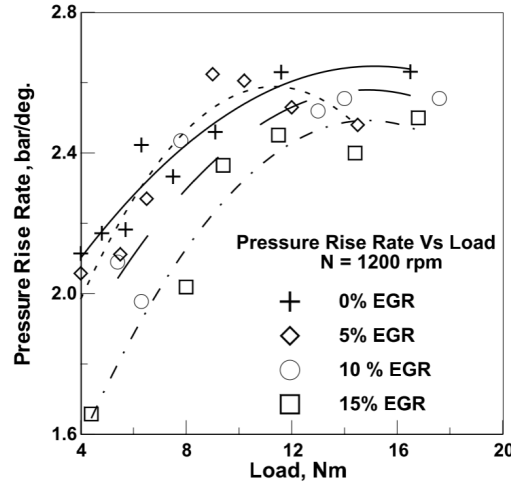
Source: *Effect of exhaust gas recirculation on some combustion characteristics of dual fuel engine, M.Selim, 2003*

The dilution effect affects to a large extent the combustion process by replacing part of the incoming fresh air with the exhaust stream, which leads to a reduction of the level of oxygen available into the intake mixture. This variance in the oxygen concentration is shown by affecting the quality of the combustion process. The dilution effect is more notorious at high EGR ratios, since the  $O_2$  replaced share increase directly as the percentage of recirculated mass flow growths.

The dilution effect promotes disturbances in ignition delays as consequence of affecting the mixture heat capacity. It also generates a notorious reduction on the peak pressure reached into the cylinder since a larger amount of  $O_2$  is replaced by  $CO_2$  and  $H_2O$ , which suppresses the combustion process and damps the pressure rise. It influences the heat release rate as well and hence, the rate of pressure rise is affected, undergoing

a reduction of the combustion noise as the EGR ratio increases at all loads [26]. Figure 2.7 shows the behaviour of the pressure rise rate with the EGR ratio at constant engine speed.

*Figure 2.7: Effect of pressure rise rate with EGR ratio*



Source: Effect of exhaust gas recirculation on some combustion characteristics of dual fuel engine, M.Selim, 2003

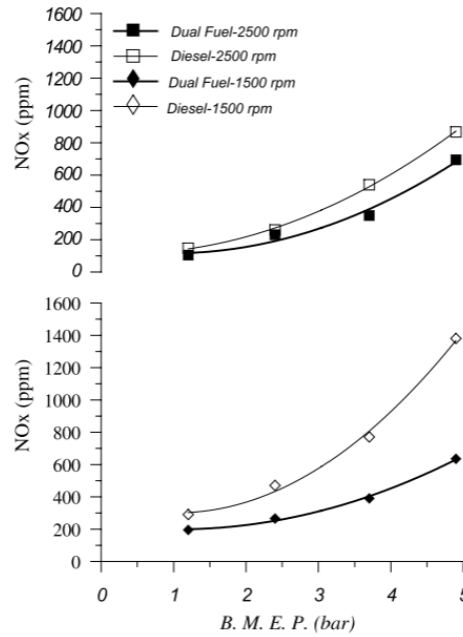
## 2.4. Emissions

The reduction of pollutants that dual-fuel operation has brought into the internal combustion engines field is one of the main features that contributed to the great success of this technology. Emissions from engines, both spark ignited, SI, and compression ignited, CI, can be divided into two subcategories; local emissions which negatively affect the environment close to the emitter, and global emissions which affect the entire planet equally. Current legislation only take local emissions into account and therefore these receive the most attention from engine developers. Global emissions, the main one being  $CO_2$ , are generally analogous to poor fuel economy and are therefore minimized in order to have a competitive product. To develop an engine is therefore necessary to find the optimum to the problem of minimizing engine fuel consumption, development cost and production costs with the constraint that the legislated emission limits need to be fulfilled. The application of dual-fuel mode significantly decreases  $NO_x$ , carbon dioxide ( $CO_2$ ) and PM emissions. However hydrocarbons (HC) and carbon monoxide (CO) emissions may increase by several times or even more than 100 times in comparison with normal diesel combustion. There appears a trade-off between  $NO_x$  and HC emissions while operating with dual-fuel mode [30].

### 2.4.1. NO<sub>x</sub> Emissions

Dual-fueling strategies in combustion ignition engines reduce  $NO_x$  emissions significantly compared to normal diesel engine operation. At low and intermediate loads the reduction is about 50% depending on engine type and conditions. Papagiannakis shows in his study [22] the trend of  $NO_x$  in DF compared with Diesel, illustrated in figure 2.8.

*Figure 2.8: Variation of nitric oxide under normal Diesel and dual-fuel operation VS load at 1500 and 2500 rpm engine speed.*



*Source: Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas, Papagiannakis, R. G. Hountalas, D. T., Energy Conversion and Management, 2004.*

The  $NO_x$  reduction is primarily due to the lower combustion temperatures, resulting from the slower flame speed of natural gas. Other studies [12] report  $NO_x$  formed during the combustion of the pilot constitutes most of the total  $NO_x$  emissions of DF combustion. Therefore, it is the type and quantity of pilot fuel that significantly affects final exhaust  $NO_x$  emissions during DF operation.

In compression ignition operation, the premixed fuel-air mixture is compressed to high temperatures during the pilot ignition delay period just prior to combustion, which accelerates the  $NO_x$  formation rate during combustion. Combustion of the natural gas-air mixture usually begins when the cycle is already late in the expansion stroke (where the charge is being cooled as the pressure drops) and hence,  $NO_x$  formation

rates from this second phase of combustion are lower.

Attempts to further reduce  $NO_x$  emissions have been made by advancing pilot fuel-injection timing [27], in which the pilot fuel is allowed to have more time to mix with the natural gas-air mixture in order to avoid locally rich mixtures and eventually,  $NO_x$  formation.

#### 2.4.2. UHC Emissions

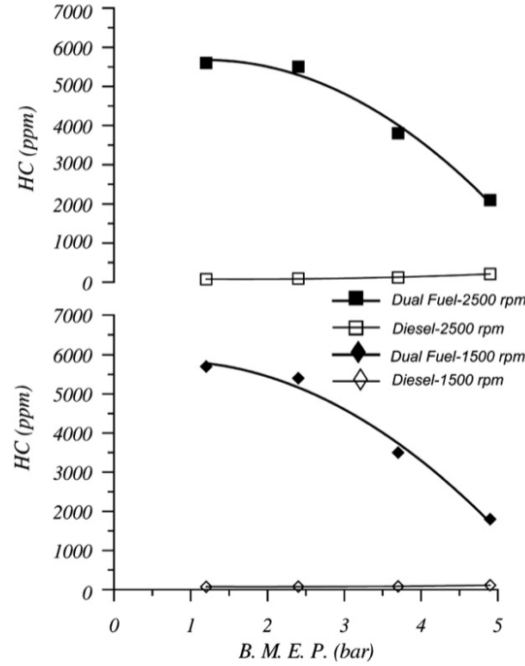
Regarding the unburned hydrocarbons emissions during dual-fuel operation, part load emissions are significantly higher than emissions found in normal CI engines at part load. The difference in emission levels is caused by unburned natural gas surviving to the exhaust because of an incomplete fuel oxidation, sudden reflect of a poor flame propagation throughout the mixture mainly within lean conditions.

T. Korakianitis, A.M. Namasivayam and R.J. Crookes state in "Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions" [15] there exists an air-fuel equivalence ratio threshold ( $\lambda$ )= 1.5 from which values above it make UHC emissions increase with decreasing overall  $\lambda$  from 5 to 1.5. Below this threshold, UHC are lower and approach those of conventional diesel operation. It is suggested by [4],[15] and [24] that at high loads UHC produced by DF are comparable with those produced by conventional CI, as the fuel-rich mixtures (still above stoichiometric) result in sufficiently high combustion temperatures to oxidize most of the fuel. As can be appreciated in figure 2.9, the UHC levels of conventional CI operation are lower than those found during DF operation for all loads, even for high loads where the emissions for DF are reduced to a large extent.

Smoke, soot and particulate emissions in DF are very low and in many cases undetectable because of the lower number of carbon atoms compared to diesel fuel (low carbon-carbon bonds and high hydrogen-to-carbon ratio), which results in lower sooting tendencies and in propitiate conditions of the mixture.



*Figure 2.9: Comparison of HC emissions for dual-fuel and conventional diesel operation as function of BMEP*



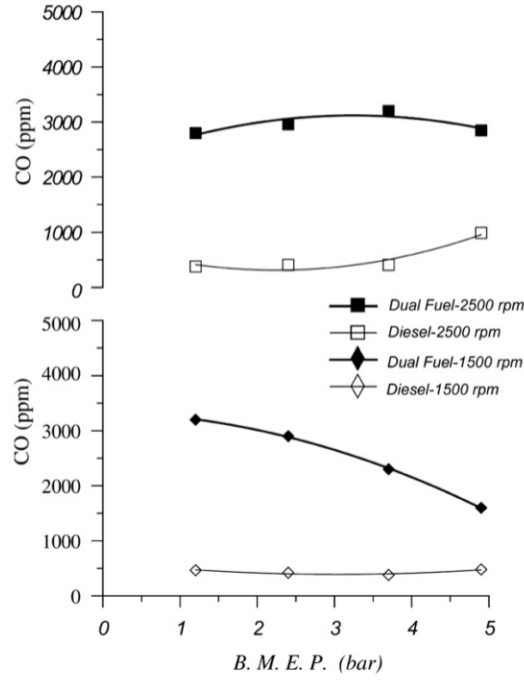
Source: *Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas*, Papagiannakis, R. G. Hountalas, D. T., *Energy Conversion and Management*, 2004.

### 2.4.3. CO emissions

CO emissions, as UHC, may be understood as indicators of the combustion quality. Large CO emissions during combustion result from low combustion temperatures and incomplete oxidation of the fuel. As Papagiannakis and Hountalas show in their analysis [22], CO emissions are significantly higher than normal CI engines for all the speed range tested throughout the load range. Other studies [4, 15] reference CO emission trends for dual-fuel operation start at a comparatively higher value than normal CI operation at low loads, and progressively approach normal diesel engine levels with decreasing air-fuel equivalence ratio, as a result of higher combustion temperatures at high loads. The comparison between CI and DF carbon monoxide emissions throughout the load range is collected in figure 2.10.

Additional solutions can be found in literature to reduce low loads emission trends on DF. For example, injection of a larger proportion of the pilot fuel contributing to the total combustion energy would provide more ignition points, thereby combustion of the natural gas-air mixture would be more complete [22, 27]. On the contrary, this further pilot injection would undeniably lead to higher diesel fuel demands.

*Figure 2.10: Comparison of CO emissions for dual-fuel and conventional diesel operation as function of BMEP*



*Source: Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas, Papagiannakis, R. G. Hountalas, D. T., Energy Conversion and Management, 2004.*

It was also found that low amounts ( $\leq 5\%$  by volume) of uncooled EGR help in increasing combustion temperatures resulting in lowered UHC and CO [1, 16, 23, 26]. This effect can be explained as the increase of charge temperature offsets the dilution effect of EGR at low loads. On the contrary, EGR substitution rates of more than 5% of the intake air during DF combustion would result in excessively high UHC and CO coupled with low NO<sub>x</sub> emissions [26].

### 3. Dual-Fuel & Exhaust Gas Recirculation Researches

The reduction in volumetric efficiency due to natural-gas injection into the intake manifold; the higher stoichiometric air/fuel ratio (17.2) above SI (14.7) and CI (14.6); and the higher  $\lambda$  at which dual-fuel engines run, could cause, under certain conditions, lower energy production than those considering gasoline and diesel as main fuel.

EGR influence into dual-fuel operation has been previously studied and analyzed in several researches [1, 2, 15]. In them, it has been proved the need of properly adjusting the recirculation rate for every working condition. However, some limitations should be carefully established when setting the EGR parameters since the combustion process and exhaust emissions are greatly influenced by slight changes on the recirculation stream. The utilization of EGR rates above a maximum value may result in misfire and erratic engine operation. For this reason, the addition of hydrogen gas as solution to increase this EGR threshold is also considered in some other studies [17].

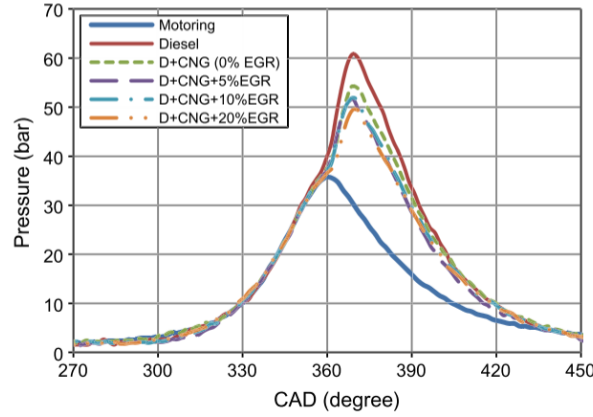
M.M. Abdelaal and A.H. Hegab in their research [2] made use of a naturally aspirated single-cylinder DI diesel engine to contrast dual-fuel operation under different load % and EGR rates.

As illustrated in figure 3.1, during part loads, conventional diesel mode exhibits higher in-cylinder pressure and earlier start of combustion than dual-fuel mode, which is attributed to the nature of the combustion process itself in each mode (heterogeneous mixture in CI combustion vs premixed combustion in DF). It is clear that the premixed combustion in DF suffers from very lean mixture, which is reflected negatively on the combustion efficiency and, consequently, may result in lower peak value of the in-cylinder pressure.

It can be appreciated the application of EGR to dual-fuel mode decreases the cylinder pressure since a large amount of oxygen is replaced by  $CO_2$  and  $H_2O$  (dilution effect). This suppresses the combustion process and damps the pressure rise, which consequently reduces the peak pressure. Additionally, the dilution effect brought by the EGR increases the ignition delay due to the growth of the mixture heat capacity.

The very lean mixture of gaseous fuel-air at part loads and its associated poor fuel efficiency lead to lower HRR values when compared with conventional diesel operation. The presence of EGR at part loads absorbs a considerable amount of the heat release as

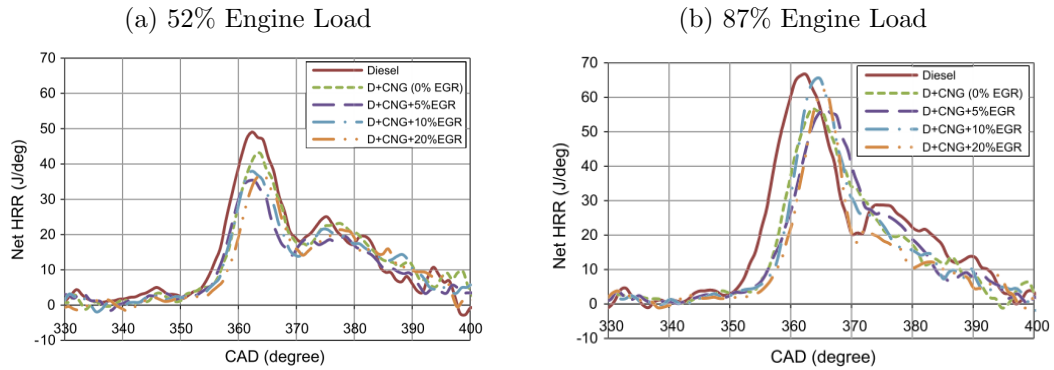
*Figure 3.1: Pressure-CAD diagram for different operating modes at 52% of the engine rated load.*



*Source: Combustion and emission characteristics of a natural gas-fueled diesel engine with EGR, M.M. Abdelaal, A.H. Hegab, 2012*

a result of the dilution effect into the mixture. In contrast, at high engine loads, larger amount of natural gas is used in dual-fuel mode, which stands for better fuel utilization efficiency. Under these conditions the addition of low EGR rates is registered positively affecting the energy release due to the chemical and thermal effect. The EGR influence along with the load % effect for both diesel and dual-fuel operation are represented in figure 3.2.

*Figure 3.2: EGR & load influence into HRR*



*Source: Combustion and emission characteristics of a natural gas-fueled diesel engine with EGR, M.M. Abdelaal, A.H. Hegab, 2012*

Regarding the thermal efficiency, M.M. Abdelaal states dual-fuel suffers from lower break thermal efficiency at part loads compared with conventional diesel mode. This is again because of the very lean mixture under those conditions and its eventual poor

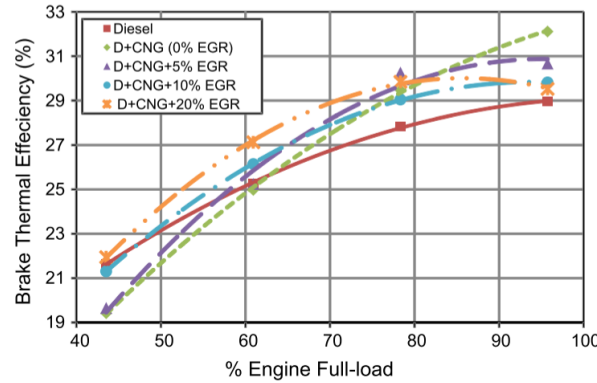
fuel efficiency (which improves as the load increases).

The very fast burning rate of the natural gas along with the improved fuel economy at high loads result in producing higher power output from the dual-fuel combustion at high load conditions compared with diesel combustion under similar conditions.

Besides, adversely affecting the brake thermal efficiency on diesel operation, the high peak of HRR associated to diesel combustion increases the radiative heat loss to the cylinder walls and to the formed exhaust.

As can be seen in figure 3.3, the utilization of a low percentage of EGR of 5% causes almost no change in the brake thermal efficiency at low loads but it brings a slight improvement at medium loads attributed to the reburning of some hydrocarbons contained in the recirculated stream. On the other hand, high EGR percentages (10-20%) lead to considerable improvements in the brake thermal efficiency at part and medium loads, because a larger amount of active radicals and unburned hydrocarbons are admitted into the cylinder. At high loads the increase of the gaseous fuel supplied together with high EGR rates reduce substantially the oxygen available for combustion, resulting in a deterioration of the combustion process aggravated as the EGR presence goes further.

*Figure 3.3: EGR & load influence into Brake Thermal Efficiency*



*Source: Combustion and emission characteristics of a natural gas-fueled diesel engine with EGR, M.M. Abdelaal, A.H. Hegab, 2012*

T.Korakianitis [15] analyzed the differences between natural-gas operation engines and conventional SI and CI modes along with the influence of EGR systems and hydrogen addition into DF engines. The study emphasizes in the importance of knocking phenomena during dual-fuel mode. Following T. Korakianitis criteria, the rate of energy released in dual-fuel engines experiences a sharp reduction when compared with conventional CI operation (represented in figure 2.5), which makes dual-fuel operation more likely to suffer from knock mostly under high loads.

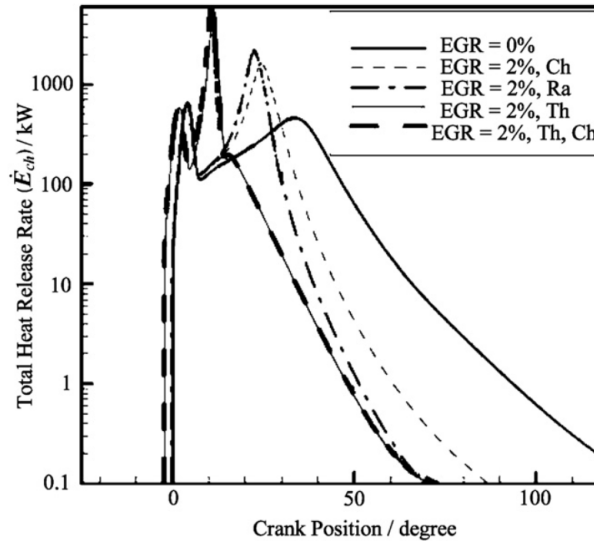
In his research it was found that the addition of low percentages of uncooled EGR

to DF engines improve low to intermediate load combustion, in addition to reducing knocking tendencies in dual-fuel engines. However, other works [26] conclude that relatively low EGR levels (of the order of 5% by mass substitution) result in increased rates of pressure rise (figure 2.7).

An increase in computed peak heat release rate (as well as to shorter combustion duration) with a 2% EGR rate compared with conventional dual-fuel operation is represented in figure 3.4, in which is appreciated the thermal (Th), chemical (Ch) and radical (Ra) effects of EGR on the heat release rate.

At high loads, EGR alleviates knocking as it dilutes the intake mixture, reducing the oxygen concentration and slowing combustion. This trend occurs due to the fact that the effect of EGR diluting the charge is significantly greater than its heating at high loads, while the reverse is true at low loads.

*Figure 3.4: Comparison of computed rate of energy of the working fluid plots ("heat release rates") for different EGR components during dual-fuel operation*

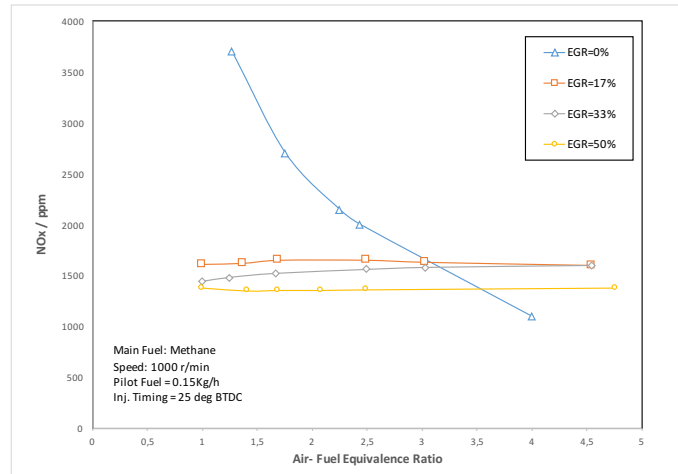


Source: Pirouzpanah V, Khoshbakhti Saray R, Sohrabi A, Niaei A. Comparison of thermal and radical effects of EGR gases on combustion process in dual fuel engines at part loads. *Energy Conversion and Management* July 2007;48 (7):1909-18

G.H. Abd-Alla [2] meanwhile carried out an study conducted on a high speed indirect injection (Ricardo-E6) dual-fuel engine and was pointed to analyze the effects of exhaust gas recirculation on the dual-fuel combustion and its emissions.

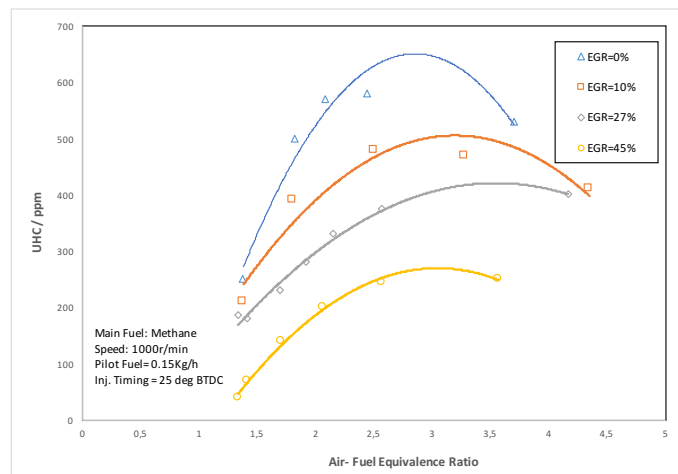
The main resolution was found that at EGR rates of about 60%, large reductions of NO<sub>x</sub> were obtained, together with a reduction of soot emissions, while a great increase of UHC was detected as can be appreciated in figures 3.5 and 3.6. However, it is also stated the use of EGR incurs some penalties, including increased specific fuel consumption and decreased heat transfer from the cylinder contents to the surrounding surface as well as affect adversely the lubricating oil quality and engine durability.

*Figure 3.5: Variations of NO<sub>x</sub> concentration with total excess-air ratio for different values of EGR ratios for methane*



*Adapted from: Effects of diluent admissions and intake air temperature in exhaust gas recirculation on the emissions of an indirect injection dual fuel engine, G.H. Abd-Alla, H.A. Soliman, O.A. Badr, M.F. Abd-Rabbo, 2001*

*Figure 3.6: Variations of unburned hydrocarbon concentration with total excess-air ratio for different values of EGR ratios for methane*



*Adapted from: Effects of diluent admissions and intake air temperature in exhaust gas recirculation on the emissions of an indirect injection dual fuel engine, G.H. Abd-Alla, H.A. Soliman, O.A. Badr, M.F. Abd-Rabbo, 2001*

## 4. RESEARCH EQUIPMENT

### 4.1. Engine

The engine utilized during this research stands for a high speed heavy-duty engine based on AGCO Power 84 CTA originally with six on-line cylinder distribution committed to the agriculture sector. For this master thesis and further researches carried out, it was adapted to one cylinder engine with the aim of reducing fuel consumption and with the hope of getting data reliable enough to extrapolate to full size engine. It is worthy to mention that, due to the one cylinder arrangement, the pressure pulses registered in the exhaust manifold are much more uneven compared to six cylinder operation where combustion takes place every  $120^\circ$ .

In order to damp the aforementioned pressure pulses, a 8 liters reservoir was located into the exhaust manifold.

*Figure 4.1: Single Cylinder Research Engine*





The table 4.1 shows the engine specifications.

*Table 4.1: Research Engine Specifications*

Parameter	Value	Unit
Engine Type	AGCO Power 84 CTA 4V	-
Number of Cylinders	6 (1 in use)	-
Cylinder Bore	111	mm
Stroke	145	mm
Compression Ratio	17:1	-
Fuel Type	Dual-Fuel (Diesel + Methane)	-

*Adapted from: Exhaust Gas Recirculation Study on Dual-Fuel Methane Combustion. Alberto Murillo Hernández, 2015*

An important detail regarding the disposal is that the operative cylinder has been chosen to be the closest to the dyno (electric motor working as dynamo-meter with a frequency converter) with the purpose of having the minimum momentum as a result of the internal forces created by the piston movement.

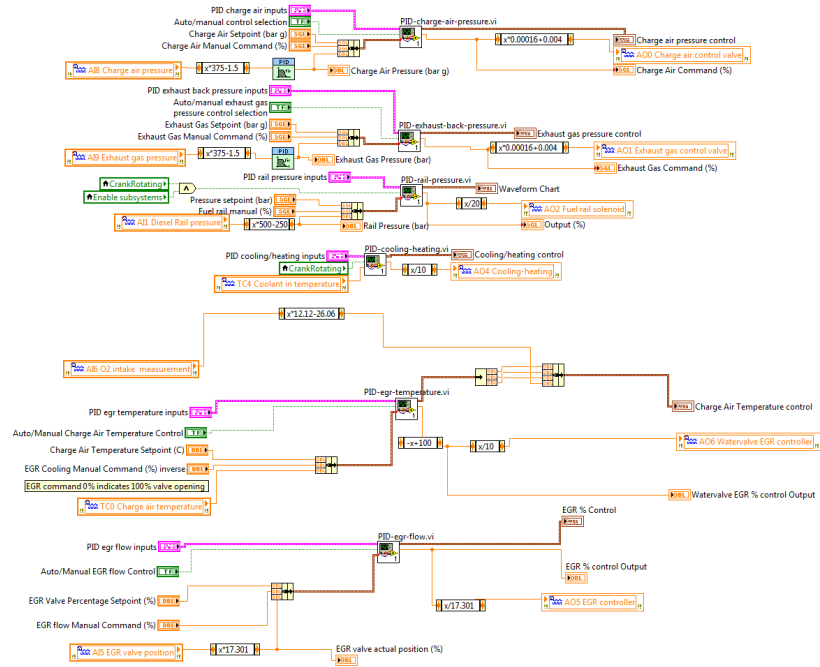
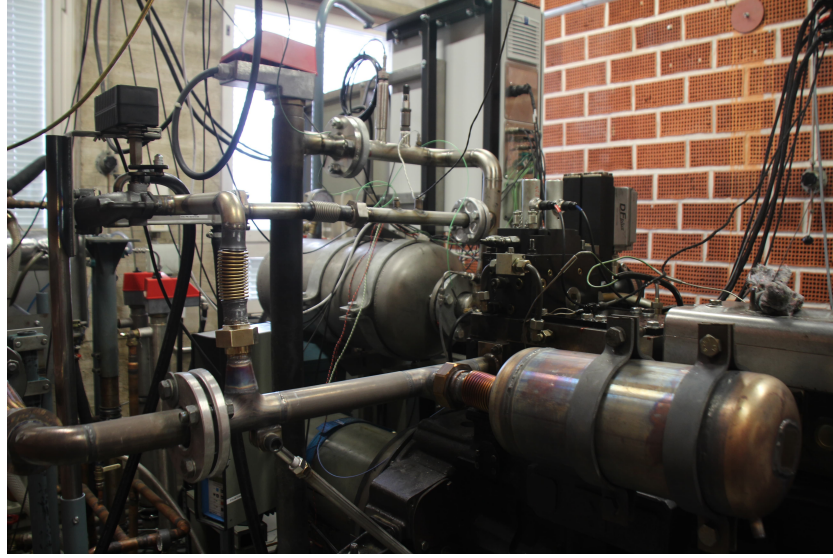
With respect to the EGR system utilized, it has been made by welding of both stainless steel and black steel pipes. Additional flex-pipes have been located into the system with the aim of withstanding and safeguard the proper functionality of the rest of the elements. As can be appreciated in the picture 4.2, an electro-valve has been placed on the EGR system allowing the variance of the recirculated stream mass flow. It has been also taken into account the high temperatures contained into the exhaust stream. For this reason and, in order to cool down the recirculated mass flow, a heat exchanger fed by the same water providing the cooling for the engine itself has been arranged.

The injection system for dual-fuel purpose in LEO I is composed by a diesel and methane subsystems separately;

The diesel system is a typical common rail layout in which only the injector of the cylinder in use is operative. In this project, the diesel rail pressure was set and controlled by means of a computerized control, and it was fixed to values close to 1000 bar.

Regarding the methane injection, two gas injectors have been located upstreams of the intake valves, which provide the methane mass flow necessary to accomplish the dual-fuel operation. The pressure difference with the intake manifold is kept constant in a value of 2,4 bar, which should guarantee a constant volumetric flow through the injectors for all the operating points.

Figure 4.2: EGR system & LabVIEW control



With the aim to compensate the lack of super/turbocharger in LEO I, an external compressor providing the demanded charge air pressure was used, allowing boost pressures up to 8 bars. Worth mentioning that a cooling/heating system was implemented along the intake pipe allowing to fit the intake air temperature to the specified requirements.

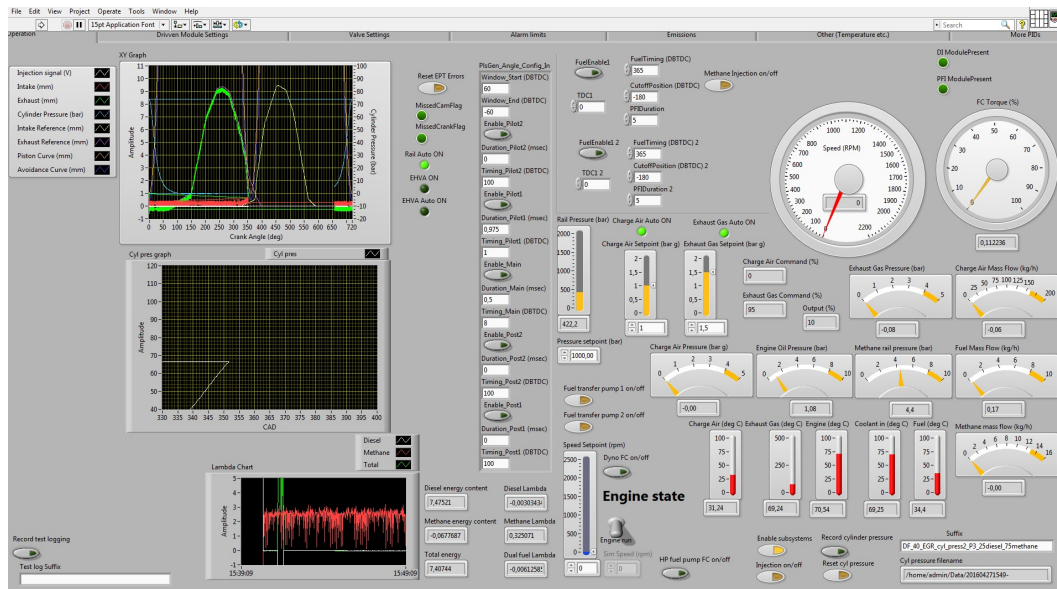
Finally, the VVT system was carried out by an electro-hydraulic valve actuator (EHVA) which permits the adjustment of the gas exchange timing by simulating the camshaft operation in a real engine.

## 4.2. Data Acquisition & Control System

### 4.2.1. Engine Control

For this MSc Thesis, LabVIEW software was used in order to perform entirely the engine control. As can be appreciated in figure 4.3, the program interface shows a set of sensors and actuators which were located throughout the engine allowing to set, in a relatively easy way, the desired conditions for every case of interest for the user. All sensors were connected to a series of electronic modules in charge of receiving the signals coming from the sensors, turn them into readable data for the user and finally transfer it to the computer.

Figure 4.3: LabVIEW Operation Interface



As the purpose of this MS's thesis is the study of the EGR effect during dual-fuel operation, a set of sensors and actuators were positioned to properly enable the establishment of conditions and parameters which give sense to the performance of this study.

The set of sensors and actuators utilized in this study consisted of:

$CO_2$  and  $O_2$  concentration analyzer for the recirculated stream (which matches with the concentration into the exhaust manifold stream).

Temperature sensor for the recirculated stream.

Intake air temperature sensor.

Valve controlling EGR cooling water flow.

Electro-valve controlling the EGR rate.

#### 4.2.2. Emission Measurements

The exhaust emissions were gathered by different modules each of them in charge of registering one different emission type. The emission analyzers require of a calibration stage to be done prior the measurements in order to set zero and full scale values as well as to avoid the disturbances produced by deviations on the surrounding air conditions which may affect the device sensitivity.

The complete system counts on long chain hydrocarbon,  $NO_x$ ,  $CO_2$ ,  $O_2$  modules, besides an additional  $CH_4$  module installed on purpose for this MS'c thesis to collect methane exhaust emissions.

*Figure 4.4: Emission Analyzer Modules*



### 4.2.3. EGR Rate Measurement

The EGR rate in this thesis is defined as the ratio between the  $CO_2$  concentration on the intake and exhaust streams, as illustrated in equation 4.1. In order to get the  $CO_2$  concentration on the air mass flow incoming to the cylinder (EGR stream + atmospheric air) it was necessary the installation of a  $CO_2$  analyzer into the intake manifold, while as already mentioned, the emission machine was in charge of collecting the  $CO_2$  concentration on the exhaust stream among the rest of components.

$$EGR(\%) = 100 \cdot \frac{[CO_2]_{inlet}}{[CO_2]_{exhaust}} \quad (4.1)$$

*Figure 4.5: Intake  $CO_2$  and  $O_2$  Analyzer*



### 4.3. Processing of Results

Processing and gathering of all the experimental results from the tests have been carried out with Microsoft Excel and Matlab. Processing of parameters such as lambdas, temperatures and oxygen contents were implemented by using Excel for its simplicity and versatility, while for those parameters requiring large amount of data for its calculation such as cylinder pressures, heat release rates or cumulative heat releases, matlab was chosen due to its higher performance compared to excel regarding processing



matrix, structure and cell data. Besides, matlab provides better noise attenuation, which is a required characteristic owing to the highly sensitive data to process.

### 4.3.1. Post-Processing

#### HEAT RELEASE

Cylinder pressure versus crank angle data over the compression and expansion strokes can be used to obtain quantitative information on the progress of combustion, such as Heat Release Rate (HRR) curves.

Prior to the heat release calculation, an averaged pressure curve from different cycles (5 segs at 1500 rpm = 70 cycles) was obtained. Heywood [11] includes a demonstration of the heat release expression assuming the contents of the cylinder can be modelled as an ideal gas.

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt} \quad (4.2)$$

Where  $Q_n$  is the net heat release rate (J/°CA)

$\gamma$  is the isentropic constant (1.35)

$p$  is the cylinder pressure as a function of crank angle (Pa)

$V$  is the cylinder volume as a function of crank angle ( $m^3$ )

This expression can be expressed as well as function of the crank angle as illustrated in equation 4.3

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (4.3)$$

Derivative of cylinder pressure, as it is a discrete value function of crank angle, can be simplified as written in equation 4.4

$$\frac{dp}{dt} = \frac{dp}{d\theta} = \frac{\Delta p}{\Delta \theta} = \frac{p_{i+1} - p_{i-1}}{\theta_{i+1} - \theta_{i-1}} \quad (4.4)$$

where  $\theta$  is crank angle (°CA).

Likewise, derivative of cylinder volume can be calculated as follows:

$$\frac{dV}{dt} = \frac{dV}{d\theta} = \frac{\Delta V}{\Delta \theta} = \frac{V_{i+1} - V_{i-1}}{\theta_{i+1} - \theta_{i-1}} \quad (4.5)$$

The calculation of the cylinder volume at any crank angle position  $\theta$  is made attending to the piston area and the stroke.

$$V = V_c + \frac{\pi B^2}{4}(l + a - s) \quad (4.6)$$

where B is the cylinder bore

l is the connection road length

a is the crank radius

s is the distance between the crank axis and the piston pin axis, given by

$$s = a \cdot \cos\theta + \sqrt{l^2 - a^2 \sin^2\theta} \quad (4.7)$$

### EGR RATE

The percentage of exhaust gas recirculation is defined as the percent of the total intake mixture which is recirculated. Thus,

$$EGR(\%) = 100 \cdot \frac{\dot{m}_{egr}}{\dot{m}_{air} + \dot{m}_{fuel} + \dot{m}_{egr}} \quad (4.8)$$

where  $\dot{m}_{egr}$ ,  $\dot{m}_{air}$  and  $\dot{m}_{fuel}$  are the mass flows of the recirculated stream, air and fuel (diesel + methane) respectively.

However, when several measurements are done consecutively, an efficient equation to calculate the EGR rate can be easily build from the  $CO_2$  balance in the inlet port.

$$[CO_2]_{atm} \cdot \dot{m}_{air} + [CO_2]_{egr} \cdot \dot{m}_{egr} = [CO_2]_{inlet} \cdot (\dot{m}_{air} + \dot{m}_{egr}) \quad (4.9)$$

Hence, using equations 4.8 and 4.9, considering the  $CO_2$  concentration in the recirculated stream equals to the  $CO_2$  concentration in exhaust stream, disregarding the  $CO_2$  concentration of the atmospheric air and despising the fuel amount compared to the intake air mass flow, the equation to calculate the EGR rate remains as illustrated in equation 4.1.

### LAMBDA

The Air-fuel equivalence ratio, also known as lambda, is the ratio between the real air-fuel ratio (AFR) and the stoichiometric one for a given blend.

In the dual-fuel case, three different lambda are considered; methane lambda, diesel lambda and total lambda, being the last one of imajor importance.

Methane Lambda:

The methane lambda is defined as the relation between the amount of air and methane injected in the cylinder compared to the methane stoichiometric value.

$$\lambda_{CH_4} = \frac{AFR_{CH_4}}{AFR_{CH_4,stoich}} \quad (4.10)$$

where  $AFR_{CH_4,stoich} = 17,2$  is the stoichiometric air-fuel ratio of methane.  $AFR_{CH_4}$  is the actual air-methane ratio, given by 4.11.

$$AFR_{CH_4} = \frac{\dot{m}_{air} + \dot{m}_{air,recirculated}}{\dot{m}_{CH_4}} \quad (4.11)$$

where  $\dot{m}_{air,recirculated}$  stands for the total amount of air re-directed to the intake, and it is calculated by means of the oxygen content registered on the exhaust gases.

Diesel Lambda:

Diesel lambda, defined as the relation between the amount of air and diesel injected in the cylinder compared to the diesel stoichiometric value.

$$\lambda_{DI} = \frac{AFR_{DI}}{AFR_{DI,stoich}} \quad (4.12)$$

where  $AFR_{DI,stoich} = 14,6$  is the stoichiometric air-fuel ratio for diesel.  $AFR_{DI}$  is the actual air-diesel ratio given by 4.13.

$$AFR_{DI} = \frac{\dot{m}_{air} + \dot{m}_{air,recirculated}}{\dot{m}_{DI}} \quad (4.13)$$

Dual-Fuel Lambda:

For the calculation of the dual-fuel lambda, methane and diesel mass flows are used, as the ratio after diesel injection is searched.

Methane lambda has more importance for the analysis compared to the diesel one, but checking if this one has a similar trend can show up if the calculation has been made properly.

In this case, methane and diesel masses should be taken into account since both fuels are used for the calculation. Methane share is calculated with the following equation using the averaged mass flows of both fuels for each operating point.



$$m_{CH_4}(\%) = \frac{\dot{m}_{CH_4}}{\dot{m}_{CH_4} + \dot{m}_{DI}} \quad (4.14)$$

Dual-fuel lambda equation:

$$\lambda_{DF} = \frac{AFR_{DF}}{AFR_{CH_4,stoich} \cdot m_{CH_4\%} + AFR_{DI,stoich} \cdot (1 - m_{CH_4\%})} \quad (4.15)$$

where  $AFR_{CH_4,stoich}=17,2$  is the stoichiometric air-fuel ratio of methane.

$AFR_{DI,stoich}=14,6$  is the stoichiometric air-fuel ratio of diesel.

$AFR_{DF}$  is the actual air-fuel ratio of the methane-diesel mixture, given by 4.16.

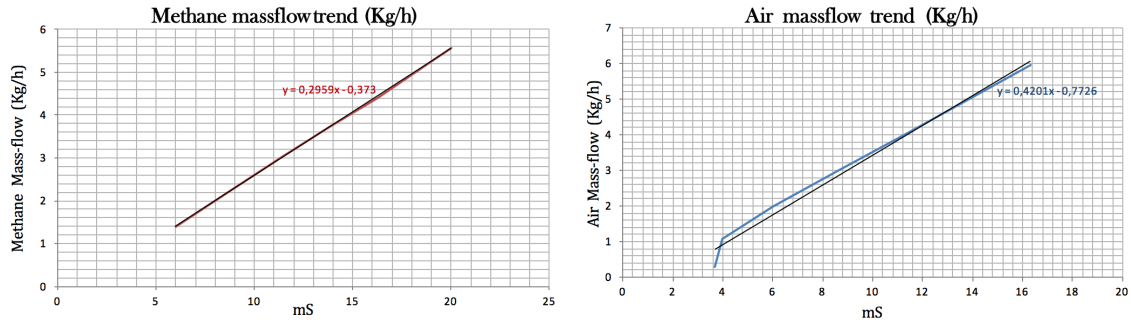
$$AFR_{DF} = \frac{\dot{m}_{air} + \dot{m}_{air,recirculated}}{\dot{m}_{CH_4} + \dot{m}_{DI}} \quad (4.16)$$

### 4.3.2. Uncertainties During Experiments

During the tests carried out in LEO I along this master thesis, some deviations were found that may influence the results up to certain point.

First of all, the methane injection system presented a problem regarding the coriolis's meter in charge of registering the methane mass flow through the injectors. The equipment was appropriate to measure mass flows according to the initial 6 cylinder disposal ( $\dot{m}_{CH_4} \in (3 - 150)$  Kg/h according to specifications) however, the aforementioned range was too high to proper measure mass flows for one single cylinder engine. The adapted solution was to analyze the trend of the injection system by testing it first with air (higher density than methane, which allowed us to get good resolution in a higher range of mass flows) and afterwards with methane itself in order to compare with the trend got by using air and check the reliability of the measurements in methane. Figure 4.6 shows the trends for air and methane applied through the coriolis's meter.

Figure 4.6: Air-Methane mass flow Trends through the Coriolis's meter



This approximation shows the coriolis's meter trend could be considered as linear for values of mass flow above 1,2 Kg/h. Hence, to carry out the tests, and only for values of  $\dot{m}_{CH_4}$  below the threshold, the equation representing the methane mass flow trough the injector is used to calculate the approximate methane amount injected.

Some deviations were found during the emission measurements since the scaling for CO and hydrocarbon modules were not high enough to register the emission values for EGR rates above 40%.

Another uncertainty regarding the emission measurement system was the inability of the hydrocarbon analyzer to properly separate UHC emissions coming from the diesel from other UHC belonging to methane. As a result, it may be possible the hydrocarbon analyzer registered values of UHC above the actual value (UHC emissions coming from  $CH_4$  were registered by the  $CH_4$  methane analyzer). Besides, the data was collected every five minutes after the parameters were set, which may compromise up to some point the accuracy of the emission data.

Finally, it must be known that the EGR control valve did not allow a proper positioning permitting to have the desired EGR values. This effect caused that no strict values of EGR rate were accomplished, finding deviations up to 2,5% from the original value.

## 5. TESTS & RESULTS

The tests carried out during this thesis were pointed to analyze the effect of the exhaust gas recirculation during dual-fuel operation on part-high loads. A test matrix for every operating point and load was developed, as can be seen in tables 5.1 and 5.2.

*Table 5.1: Test Matrix 50% Load*

	0% Substitution Rate	30% Substitution Rate	50% Substitution Rate	75% Substitution Rate
<b>0% EGR</b>				
<b>15% EGR</b>				
<b>30% EGR</b>				
<b>40% EGR</b>				
<b>50% EGR</b>				

*Table 5.2: Test Matrix 60% and 70% Load*

	0% Substitution Rate	30% Substitution Rate	50% Substitution Rate	75% Substitution Rate
<b>0% EGR</b>				
<b>15% EGR</b>				
<b>30% EGR</b>				
<b>50% EGR</b>				

Worth mentioning that the substitution rate was defined as the share of energy provided by each fuel, which differs from the shares of fuel mass flow as the methane calorific power is higher than diesel.

*Table 5.3: Methane & Diesel Calorific Power. [28]*

<b>Fuel</b>	<b>Calorific Power (MJ/Kg)</b>
Methane	55,53
Diesel	44,8

Therefore, the fuel shares were calculated as appears in the equation 5.1.

$$E_{content} = A \cdot E_{diesel} + B \cdot E_{methane} \quad (5.1)$$

Where A and B represent the shares of each fuel respectively, and  $E_{diesel}$ ,  $E_{methane}$  are calculated as follows:

$$E_{fuel} = \dot{m}_{fuel} \cdot CGV \quad (5.2)$$

Where  $\dot{m}_{fuel}$  (Kg/h) indicates the amount of fuel injected and CGV (KJ/Kg) is the gross calorific power for a specified fuel.

Several mechanical limitations restricting the engine operation range were found and taken into account while defining the operation boundaries. These are represented in table 5.4.

*Table 5.4: Mechanical Limitations*

Parameter	Value
Cylinder Pressure (bar)	220
PRR (bar/ $\theta$ )	10
Diesel Injection Timing (mS)	0,3

With respect to these limitations, the value of maximum load was set regarding the maximum cylinder pressure the engine is capable of withstanding (220 bar). Hence, based on the cylinder pressures registered after the tests, the experimental data was gathered from 50 to 70% of maximum load according to the equation 5.3.

$$Load(\%) = 100 \cdot \frac{CylinderPressurePeak}{220} \quad (5.3)$$

It must be clarified that the boundary set for the rates of pressure rise is based on conventional values utilized during prior researches and hence, no theoretical or practical studies were develop to ensure the reliability of this value.

The diesel injection utilized is a so-called peak & hold injector, whose specifications sheet restrict its duty time to a minimum of 0,3 mS.

In order to get comparable results among all tests carried out on different loads, there was necessary the establishment of a common setup which would be kept constant for every experiment.

The parameters concerned basically the injection system, in which SOI was maintained for all tests, as well as the valve timing, whose setup did not vary either along the whole experiments. Furthermore, no pilot injection was considered during the tests and the common rail pressure was also remained constant.

The injection and valve timing pre-setup as well as the engine speed may be seen on the following table.

*Table 5.5: Common Setup*

<i>Diesel SOI</i>	8° BTDC
<i>Methane SOI</i>	5° ATDC
<i>Pilot Injection</i>	-
<i>Diesel Pressure Rail</i>	1000 bar
<i>Methane Injection Pressure</i>	2,4 bar over intake air pressure
<i>Intake Valve Timing</i>	IO = 32,5° BTDC, IC = 50° ABDC
<i>Exhaust Valve Timing</i>	EO = 20° BBDC, EC = 39° ATDC
<i>Engine Speed</i>	1500 rpm

## 5.1. 50% LOAD OPERATION

Tests made on 50% load were shown in table 5.1. The experiments during 0% and 30% substitution rates were performed in a range from 0-50% EGR. However, deviations in the EGR control valve did not allow to reach the desired points for 50% and 75% SR. Therefore, the points analyzed for 75% SR were; 0%-15%-40% EGR, while for 50% SR; 0%-15%-27%-31%.

In addition, and with the aim of studying the effect on combustion caused by higher EGR rates, 75% of EGR was also set specifically for a methane share of 50%.

Due to the large amount of information gathered, both graphic and numerical data shown in this section have been chosen to be the most remarkable among the whole tests.

The conditions established regarding the fuel shares during 50% load tests are exposed in the following table:

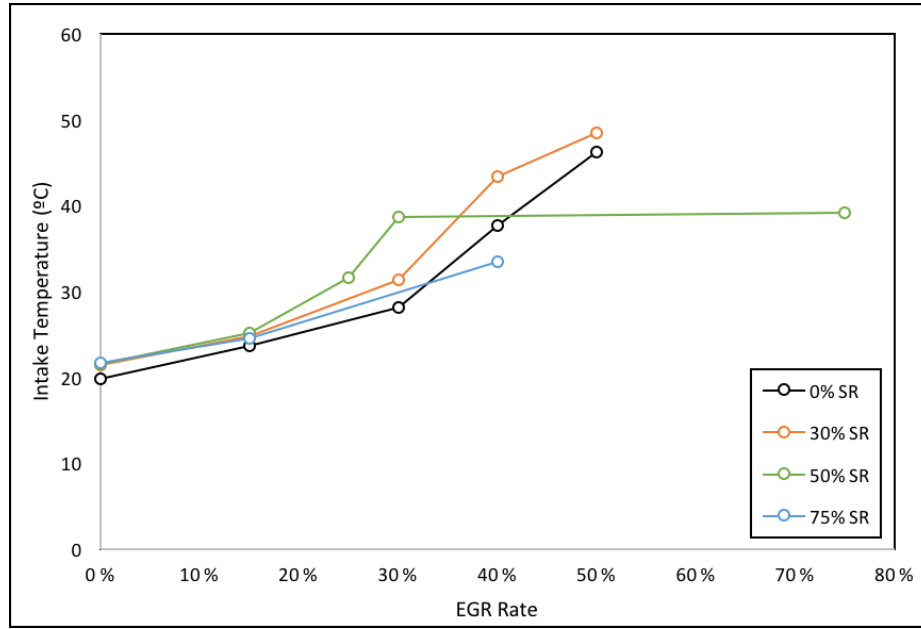
*Table 5.6: 50% Load Conditions*

	Charge Air Pressure (bar)	Back Pressure (bar)	Energy Content (KJ/s)	$\dot{m}_{\text{diesel}}$ (Kg/h)	$\dot{m}_{\text{methane}}$ (Kg/h)
<i>0% Substitution Rate</i>	1	1,5	175	4,022	0
<i>30% Substitution Rate</i>	1	1,5	175	2,97	0,96
<i>50% Substitution Rate</i>	1	1,5	175	1,966	1,59
<i>75% Substitution Rate</i>	1	1,5	175	1,048	2,386

It must be noted the back pressure was chosen to be high enough to allow the recirculated stream flowing towards the intake manifold.

Despite the installation of a heat exchanger on the EGR system, the cooling capacity of this turned to be not enough to cool down the recirculated mass flow to a expected value of 20°C. As a result, the intake blend experienced an increase of temperature, as can be appreciated in figure 5.1. Although it is not a marked effect, the intake mass flow temperature needs to be watched out since it is closely related to the appearance of knocking and ease the release of  $NO_x$  emissions.

*Figure 5.1: Charge Air Temperature VS EGR Rate*

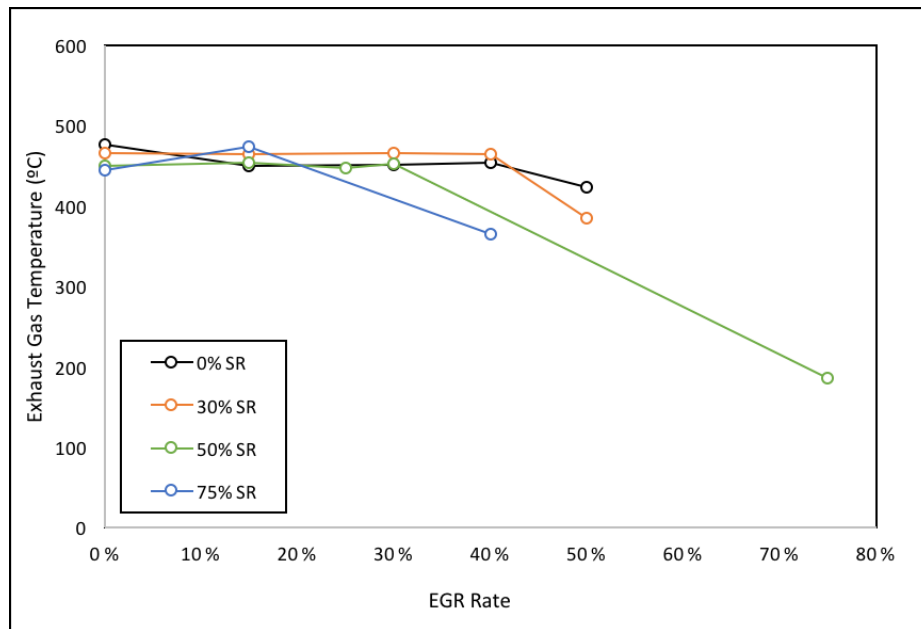


The temperature of the exhaust gases is affected as well by both EGR and variations of the substitution rate, as graphed in figures 5.2 and 5.3. As illustrated, the effect of varying the fuel share is shown to be non comparable to the effect created by different EGR rates, which is found to be more notorious while increasing the recirculated stream mass flow.

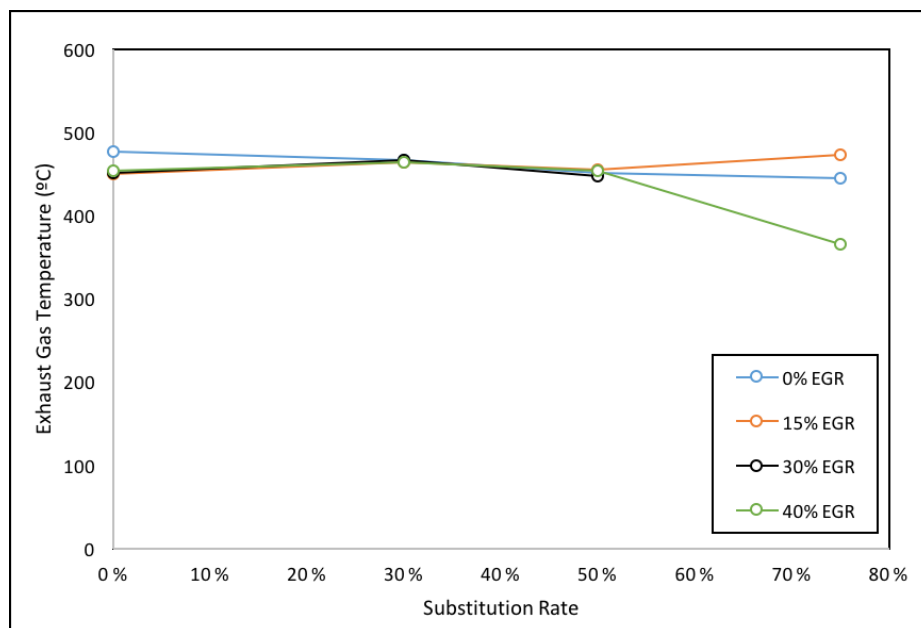
The exhaust temperature reduction may be explained as direct consequence of the dilution effect which damps the combustion and, hence, the peak temperatures achieved during the process itself.

This dilution effect is understood as a depletion on the intake mass flow oxygen content and is represented in figure 5.4. The severe  $O_2$  reduction leads to important variations on cylinder pressures (figure 5.5) and therefore it is also considered a critical parameter for the energy released during combustion as appears in figures 5.7 and 5.8.

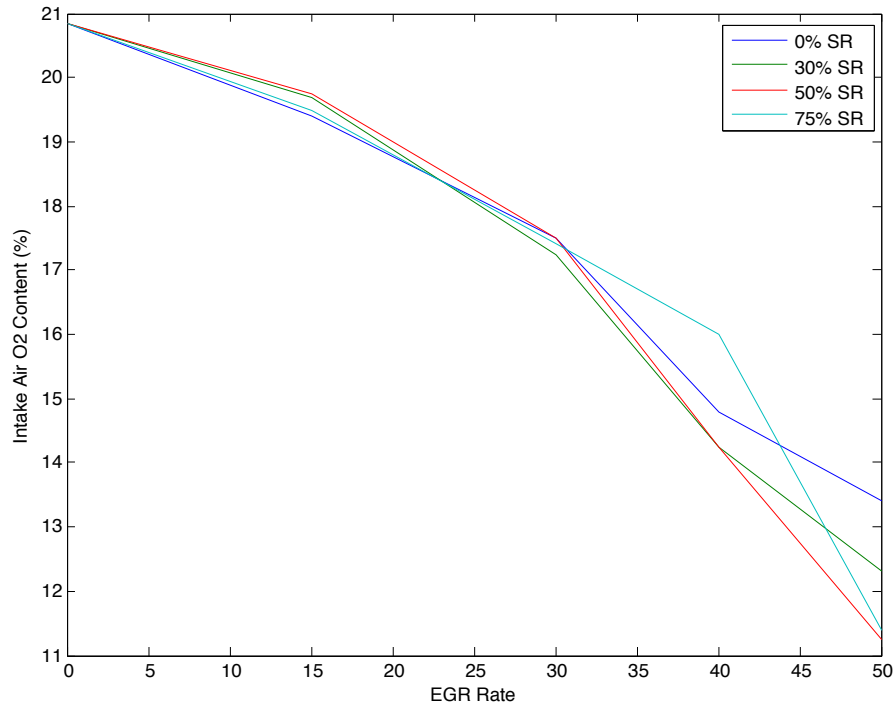
*Figure 5.2: Exhaust Gas Temperature VS EGR Rate*



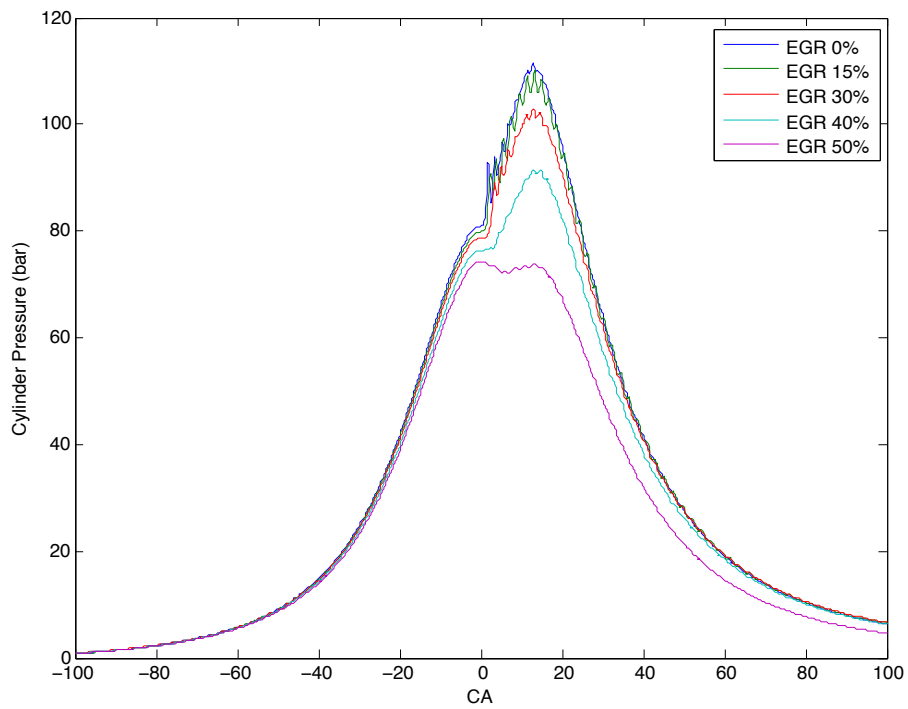
*Figure 5.3: Exhaust Gas Temperature VS Substitution Rate*



*Figure 5.4: Oxygen Content into the Intake mass flow for different EGR and Substitution Rates.*



*Figure 5.5: Cylinder Pressure for different EGR Rates at 30% Substitution Rate and 50% Load*





The figure 5.5 shows the changes experimented by the cylinder pressure while varying the EGR rate for a diesel substitution rate of 30%. As can be appreciated, for values up to 15% EGR the cylinder pressure does not experiment great deviations from the original value (0% EGR), while higher increases such as 30-40% show clear reductions on the peak pressure. At 50% EGR rate the combustion seems to be so poor that the peak pressure is found below 80 bar and appears 0,8° BTDC, which indicates that it is produced during the compression stroke, standing for a very low combustion quality.

Moreover, it is interesting to observe the pressure disturbances found on low EGR rates between 0-20° ATDC associated to the re-burning of some recirculated hydrocarbons and how this effect is narrowing as the EGR rate increases due to the growing influence of the dilution effect.

The figure 5.6 shows the cylinder pressure curves for a diesel share of 50% in which EGR rates up to 75% were established.

Analyzing the EGR 75% point, it can be appreciated how the compression stroke is again the responsible of producing the peak pressure (0,6° BTDC). Besides, in this case the increase of cylinder pressure during combustion is practically non-existent, which is a clear indicator of the absence of combustion, that is to say, at this point the engine suffers from misfiring.

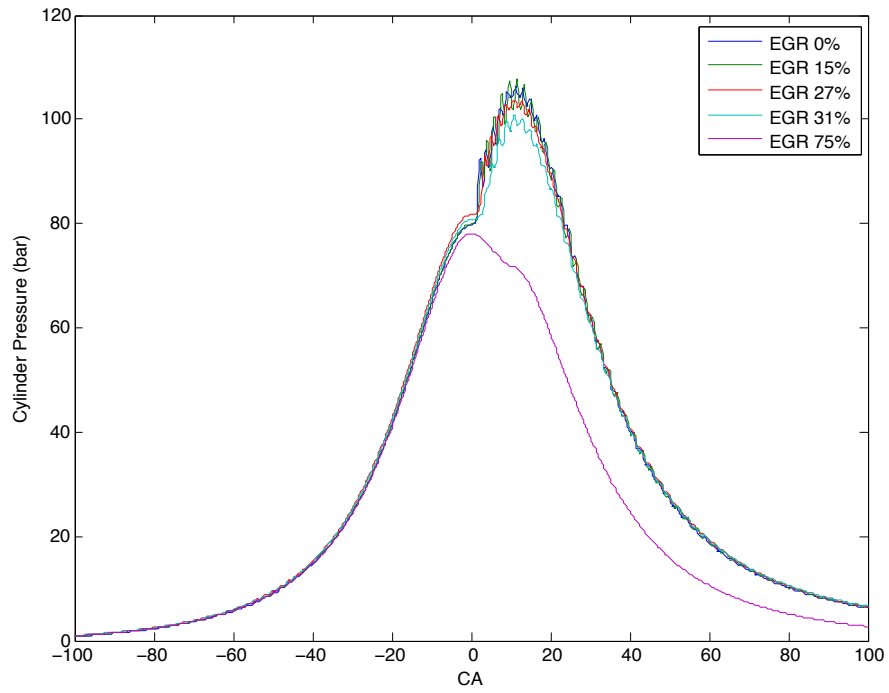
Clearly the cylinder pressure is influencing the evolution of the heat release rate as stated in the equation 4.3. Thus, lower cylinder pressures lead to lower heat release rates and hence, lower cumulative heat release during the cycle.

When looking at heat release rate and cumulative heat release curves for a methane share of 30% (figures 5.7 and 5.8), it is registered an increase of energy produced during combustion for EGR rates of 15%, a minor part of this increase may be associated to the growth of the intake temperature promoted by the recirculation (so-called thermal effect). Besides, at this low EGR percentages, the  $O_2$  level is not significantly reduced (19,89% according to figure 5.4). Both facts, along with the re-burning of some hydrocarbons, (chemical effect) influence the combustion process by causing a mild increase of the heat released and most likely a light increase in the thermal efficiency.

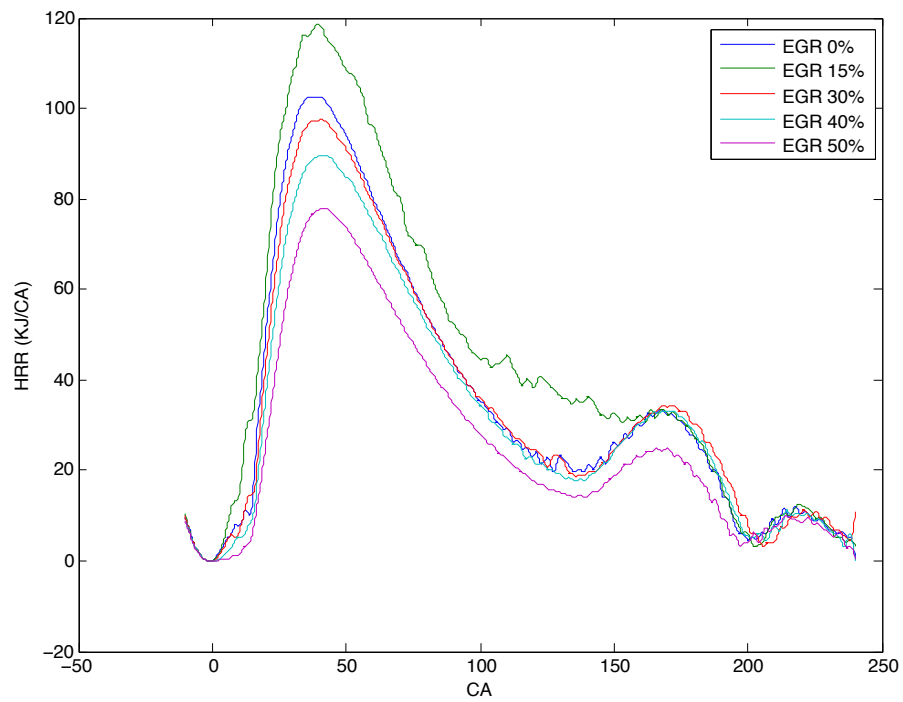
Nevertheless, very rarely these conditions bring that huge differences compared to the operation without EGR. The major part of this deviation may be attributed to the inaccuracy of the coriolis's meter to properly measure methane mass flows below 1,2 Kg/h (the table 5.6 shows a methane mass flow of 0,96 Kg/h) and therefore, the energy content data may be compromised for this point.

However the trend in both graphs is clearly showing how for higher EGR rates the energy produced is much lower, especially for those with values above of 30% of EGR.

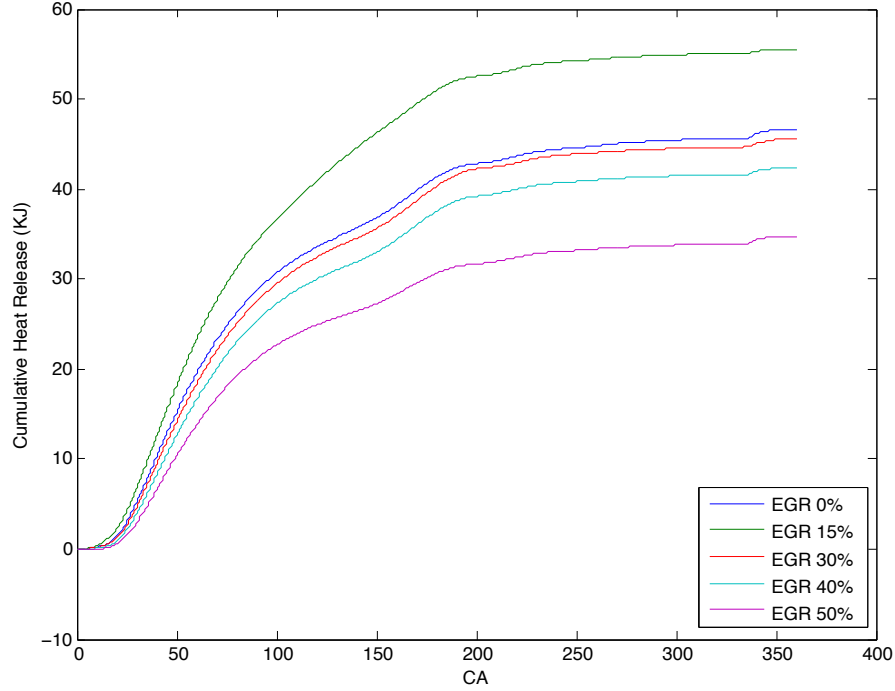
*Figure 5.6: Cylinder Pressure for different EGR Rates at 50% Substitution Rate and 50% Load*



*Figure 5.7: Heat Release Rate for different EGR Rates at 30% Substitution Rate and 50% Load*



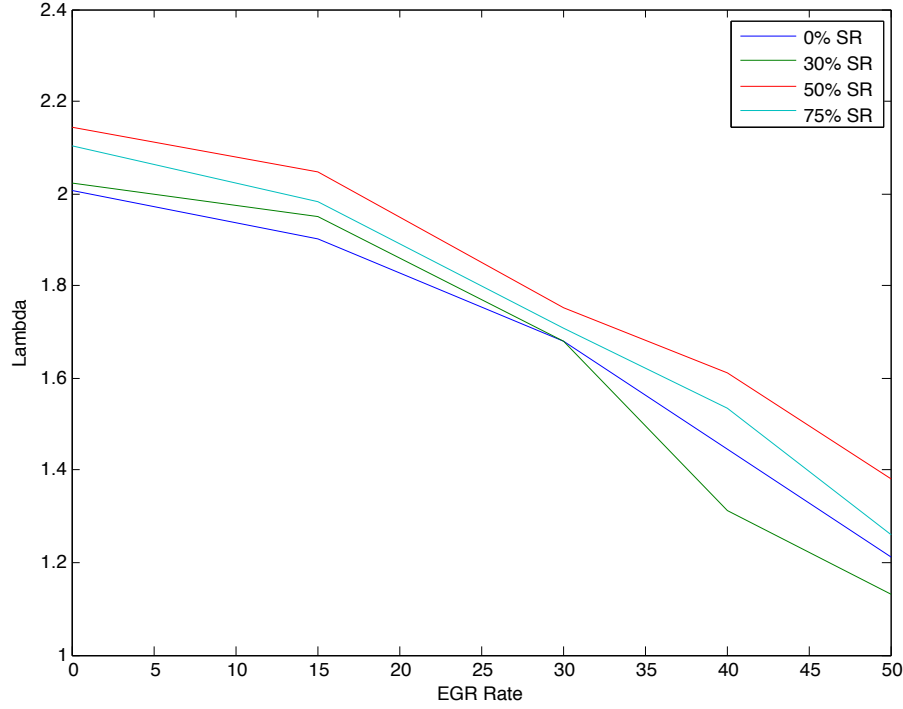
*Figure 5.8: Cumulative Heat Release for different EGR Rates at 30% Substitution Rate and 50% Load*



The main fact characterizing the conventional dual-fuel combustion is that the process is understood as a combination of premixed and diffusion flame, which enables to run during much leaner conditions than those normally possible in gas-fuelled spark ignition engines.

As mentioned, the presence of EGR promotes a lack of oxygen within the intake mixture. As the oxygen level is reduced, it is reasonable to think the dual-fuel lambda will experiment a reduction according to the evolution of the intake oxygen content (figure 5.4). This effect is shown in the figure 5.9 and was calculated as a direct application of the equation 4.15. Obviously the shares of each fuel remained unchanged within a defined substitution rate.

**Figure 5.9:** Air-Fuel Equivalence Ratio for different EGR and Substitution Rates at 50% Load



With the aim of assessing the ignition delay conduct, and due to the disturbance of the cumulative heat release data precluding to determinate the exact starting of combustion, the definition of ignition delay was established as the timing lasted to achieve a share of 5% of the total heat released during combustion. To properly proceed, the diesel SOI (8° BTDC) must be taken into account to determinate the actual ignition delay, which was calculated according to the following equation:

$$IDT = \frac{(\theta_{5\%HRR} + SOI) \cdot 60}{1500 \cdot 360} \quad (5.4)$$

Where IDT stands for the ignition delay (s).

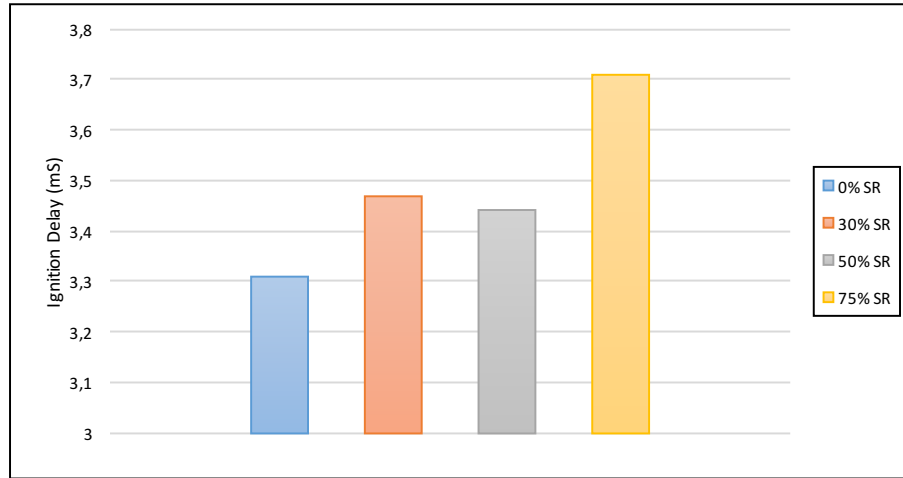
$\theta_{5\%HRR}$  is the crank angle when 5% of the total energy released is achieved.

Based on the figure 5.10, it is appreciable how the increase of methane shares within the total energy supplied leads to further ignition delays, while it remains shorter for lower substitution rates. This effect is associated mainly to the increase of the blend's heat capacity provided by the methane and, to a lesser extent, due to changes in the partial intake pressure of oxygen displaced by the gaseous fuel.

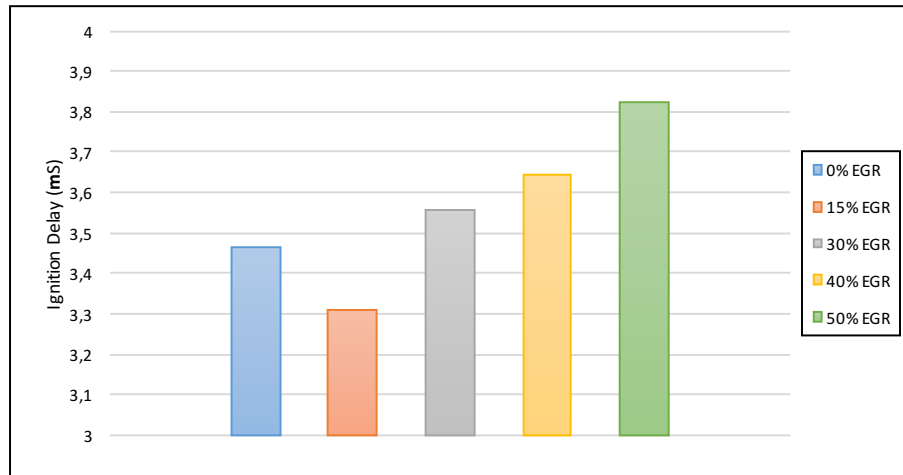
The dilution effect brought by the recirculated mass flow influences as well the lag of combustion as can be seen in 5.11. In this sense, the recirculated stream promotes, like

the increase of methane shares, growths on the mixture heat capacity forcing the delay of the ignition to increment. Evidently the phenomena acquires greater importance as the recirculated mass flow increases.

*Figure 5.10: Ignition Delay for different Substitution Rates at 0% EGR Rate and 50% Load*



*Figure 5.11: Ignition Delay for different EGR Rates at 30% Substitution Rate and 50% Load*

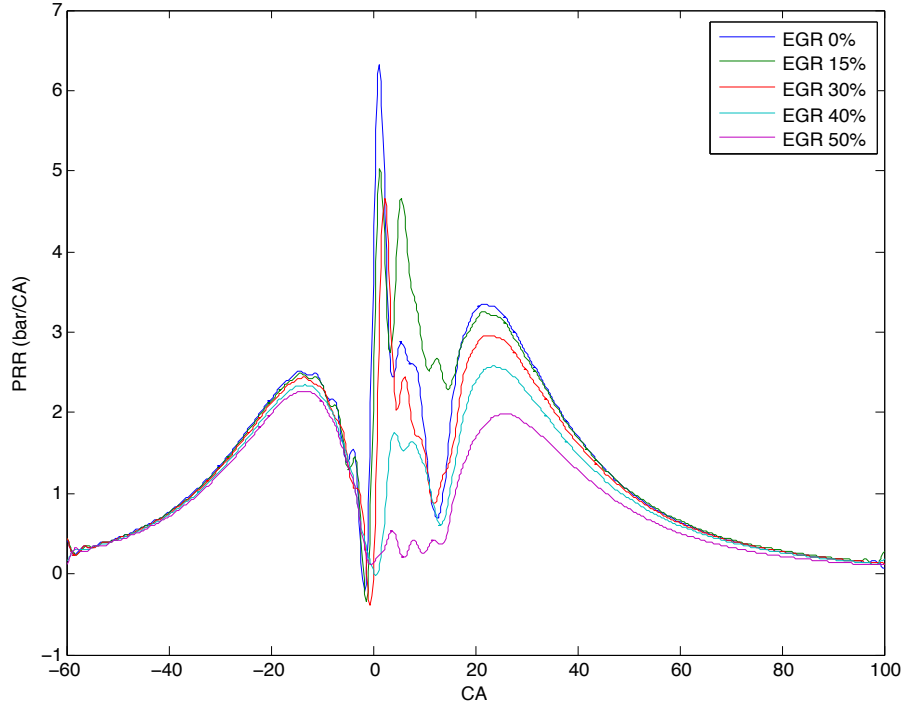


Additionally, to ensure that no knocking is appearing during the experiments and due to its narrow connection with the rate of pressure rise, it was found interesting to analyze the evolution of the aforementioned parameter in order to double-check that the magnitude of the values registered did not overtake the limitation established in table 5.4.

It is possible to appreciate that the critical values of PRR appear very close to TDC for all EGR rates, finding their maximum value barely higher than 6 bar/°CA

(the actual values could go slightly above due to the filtering). This effect appears as a result of the high pressure difference arisen from the starting of combustion.

**Figure 5.12:** *Pressure Rise Rate for different EGR Rates at 30% Substitution Rate and 50% Load*



Aside the energetic point of view, one of the prime objectives of this thesis regards the emission produced during different EGR operations. Especially, it was considered of great importance to find out how  $NO_x$  emission react to the dilution effect promoted by partial recirculation of exhaust gases since they are the main responsible of harmful environmental effects such as photochemical smog and must be reduced to a large degree.

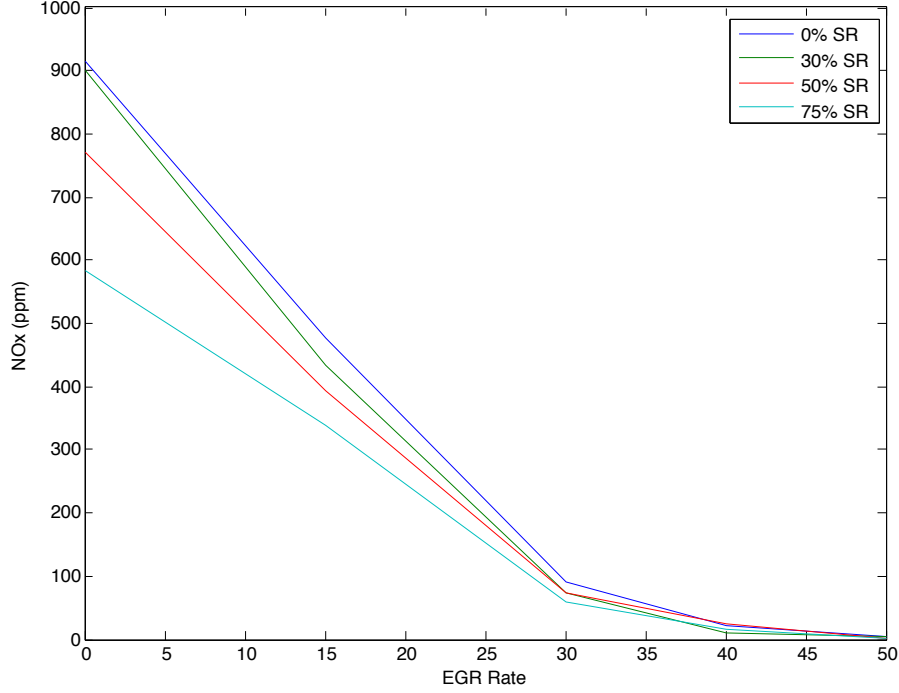
The thermal effect, based on the increase of the intake blend's temperature as a consequence of the recirculation of high enthalpy gases towards the intake manifold, along with the dilution effect, responsible of dampen the combustion by reducing the oxygen content available for the oxidation of the fuel are the main facts characterizing the reduction of  $NO_x$ .

While the first one promotes the growth of the nitrogen oxides by the enhancement of the intake mass flow temperatures, the second effect provoke a most remarkable impact in favour of the  $NO_x$  reduction due to the inability of the chemical process itself to fully oxidize the combustible leading to lower combustion temperatures as well as poor combustion quality.

This  $NO_x$  reduction has been favored during this research due to the location of a heat

exchanger into the EGR system which cooled down to a large degree the recirculated gases.

*Figure 5.13: NO<sub>x</sub> emission for different EGR and Substitution Rates*



With regards to carbon monoxide emissions, it can be appreciated that, unlike  $NO_x$ , the higher the substitution rate utilized, the greater the CO emissions registered. Obviously this effect is supported by the different fuel composition of diesel and methane. It is also notable that, while the presence of EGR is encouraging the nitrogen oxides depletion, it also gives way to the rise of carbon monoxide emissions as consequence of a poor fuel oxidation.

The figure 5.15 shows a very similar scheme while analyzing the methane emissions. In this sense, drastic growths of emissions appear when the EGR percentage reaches a value in between 30-40% (depending on the diesel share). It should be noted that for 0% SR, the total amount of fuel is provided by diesel, therefore, no  $CH_4$  is released to the ambient.

Figure 5.14: CO emission for different EGR and Substitution Rates

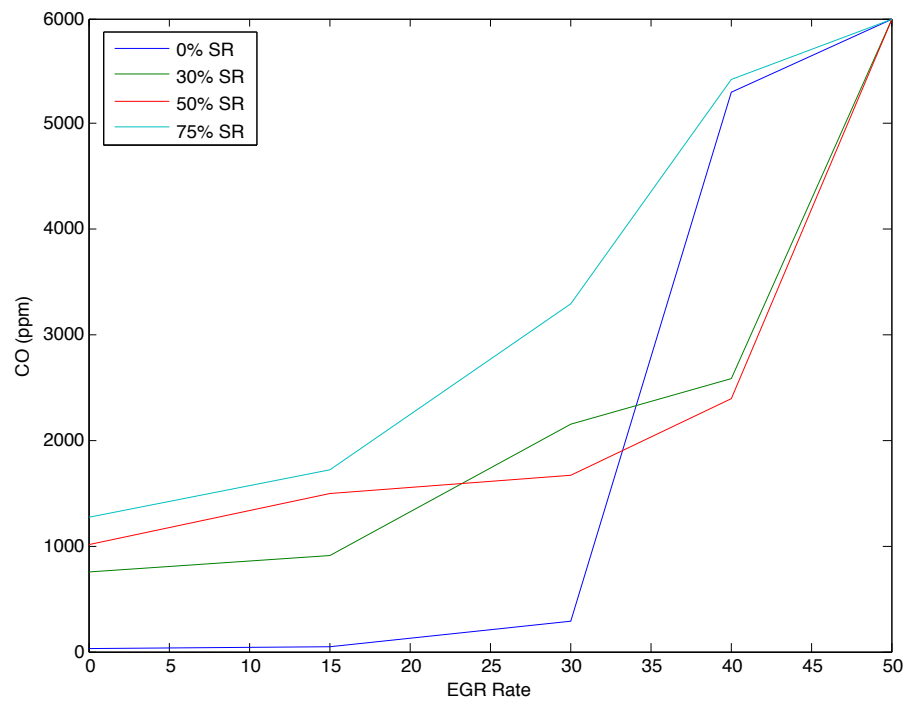
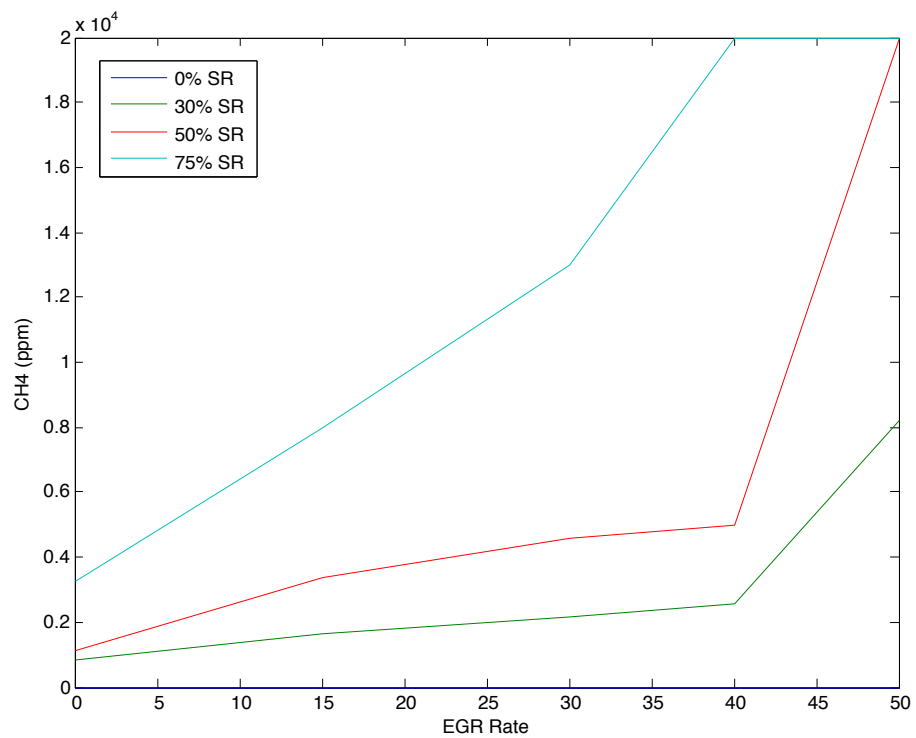


Figure 5.15: CH<sub>4</sub> emission for different EGR and Substitution Rates





## 5.2. 60% LOAD OPERATION

The tests made during the 60% load are shown in table 5.7. In this case, as well as for 70% load, the EGR percentages were chosen to be as appears in table 5.2.

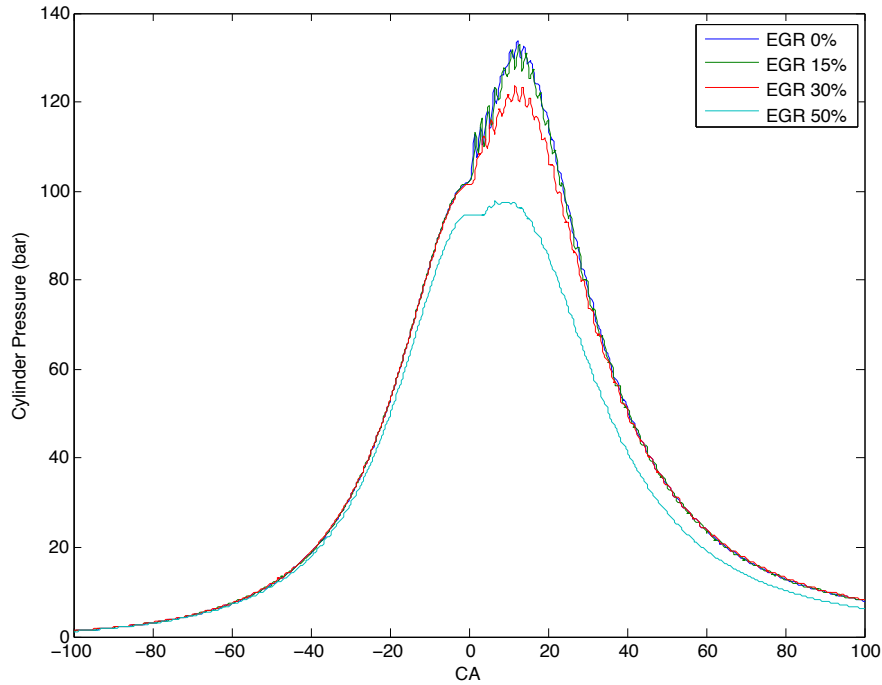
Like for the other load experiments, both graphic and numerical data shown in this section have been chosen to be the most remarkable among the whole experiments carried out.

Due to the similarity found with the results obtained on 50% load in most of the cases, figures will be swiftly commented highlighting the most particular effects when applicable.

*Table 5.7: 60% Load Conditions*

	Charge Air Pressure (bar)	Back Pressure (bar)	Energy Content (KJ/s)	$\dot{m}_{\text{diesel}}$ (Kg/h)	$\dot{m}_{\text{methane}}$ (Kg/h)
0% Substitution Rate	1,5	2	220	5,11	0
30% Substitution Rate	1,5	2	220	3,631	1,2
50% Substitution Rate	1,5	2	220	2,65	2
75% Substitution Rate	1,5	2	220	1,223	3

*Figure 5.16: Cylinder Pressure for different EGR Rates at 50% Substitution Rate and 60% Load*

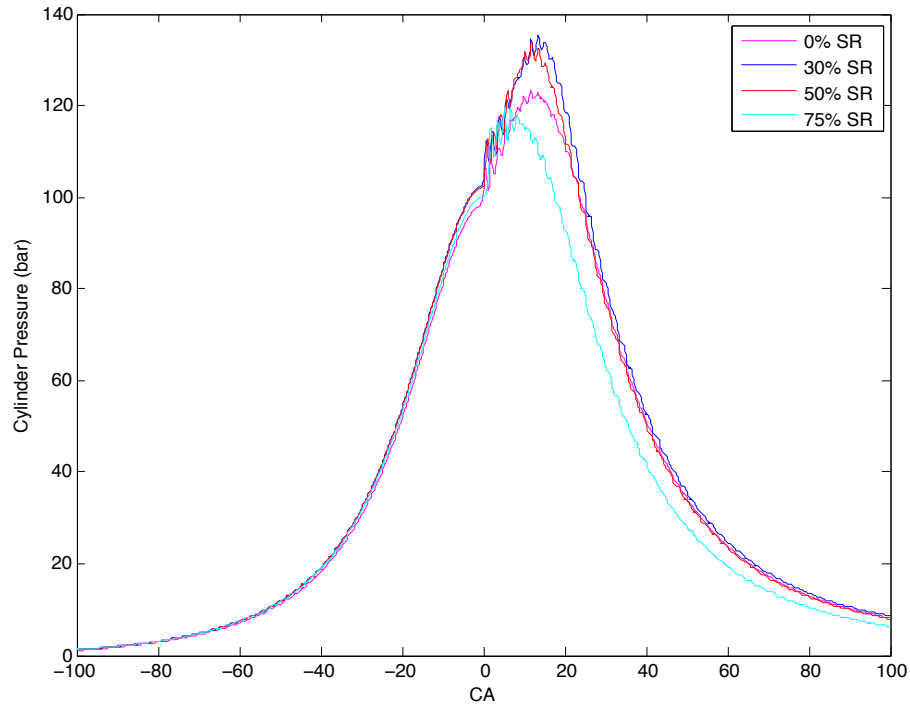


In this section, it is represented the effects produced by the variance of the substitution rates in both cylinder pressures and released energy. The understanding of the cylinder pressure trend represented in figure 5.17 has its origin in the the definition of the

dual-fuel combustion. As explained by G.A. Karim [13], a general examination of the dual-fuel combustion process in a premixed dual-fuel engine could be made through the apparent heat release rate curve, which may be considered to be made up of essentially the overlapping of three major components: the first understood as the contribution produced by the pilot injection. The second standing for the combustion of the gaseous fuel in the immediate vicinity of the pilot. And the third regarding the subsequent flame propagation within the overall lean mixture.

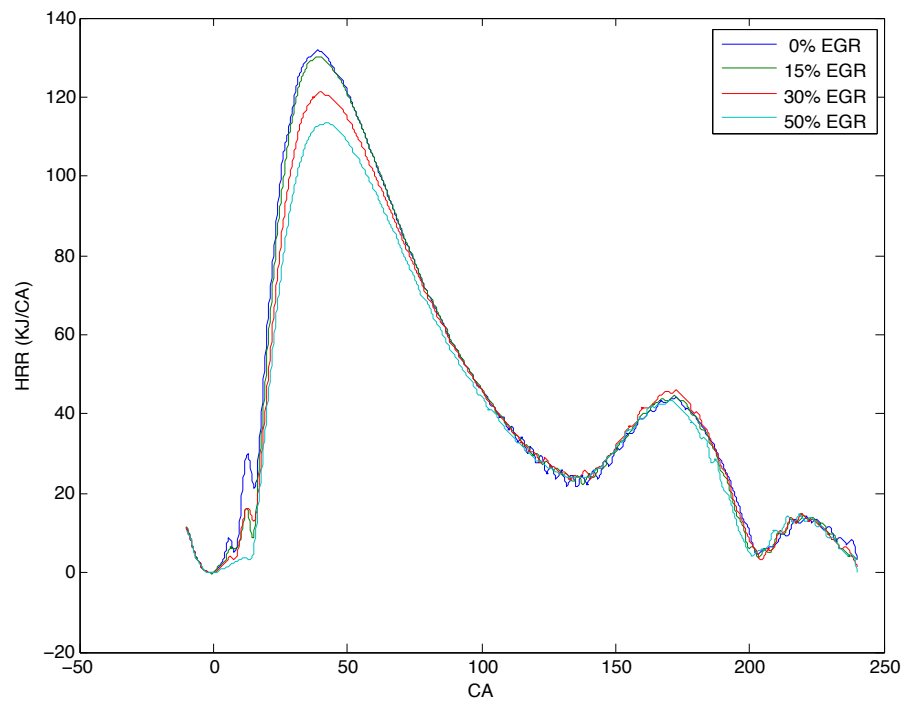
In this sense, and visualizing the cylinder pressure curves for each diesel share, it is clear how the three phases aforementioned bring great enhancements while treating substitution rates up to 50%, mainly because of the higher calorific power owned by methane. However, this beneficial effect disappears when higher methane shares are analyzed. This great decrease in the cylinder pressure occurs owing to, first of all, the large reduction of the contribution provided by the diesel injection (as reduction of the diesel share). Besides, since the dual-fuel  $\lambda$  for 0% EGR is slightly higher than 2, it is very likely that the diffusion flame speed during combustion could be reduced to some extent, leading to curves of cylinder pressures with lower peaks and flatter slopes.

*Figure 5.17: Cylinder Pressure for different Substitution Rates at 0% EGR Rate and 60% Load*

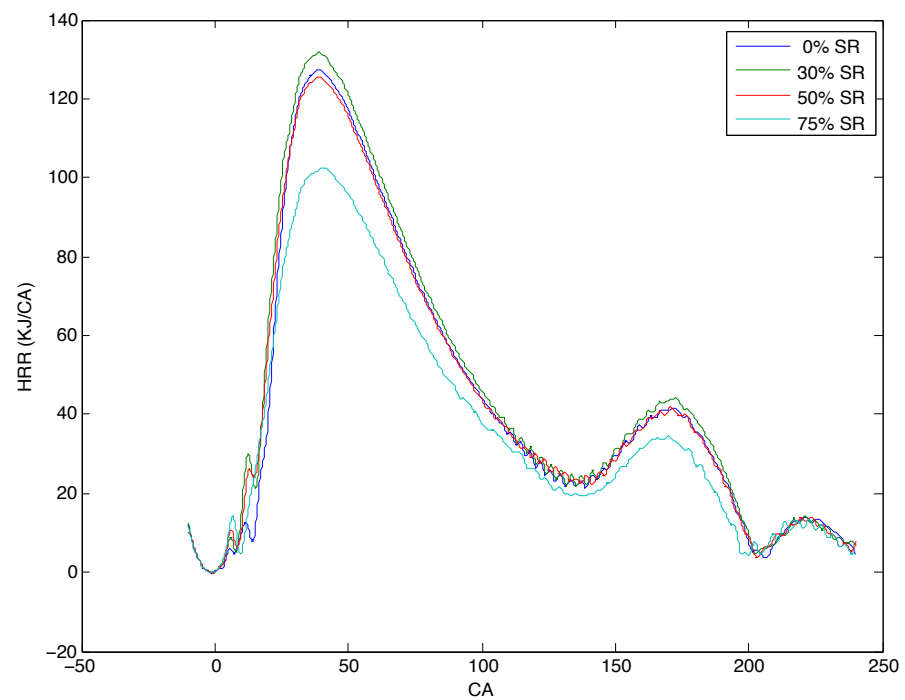


This results guide us to the rational thought of considering the possibility to obtain slight enhancements of the energy released for diesel shares of 70 and 50%, as well as its logical reduction brought by the utilization of excessively low diesel substitution rates. These results can be clearly seen in figures 5.19 and 5.21

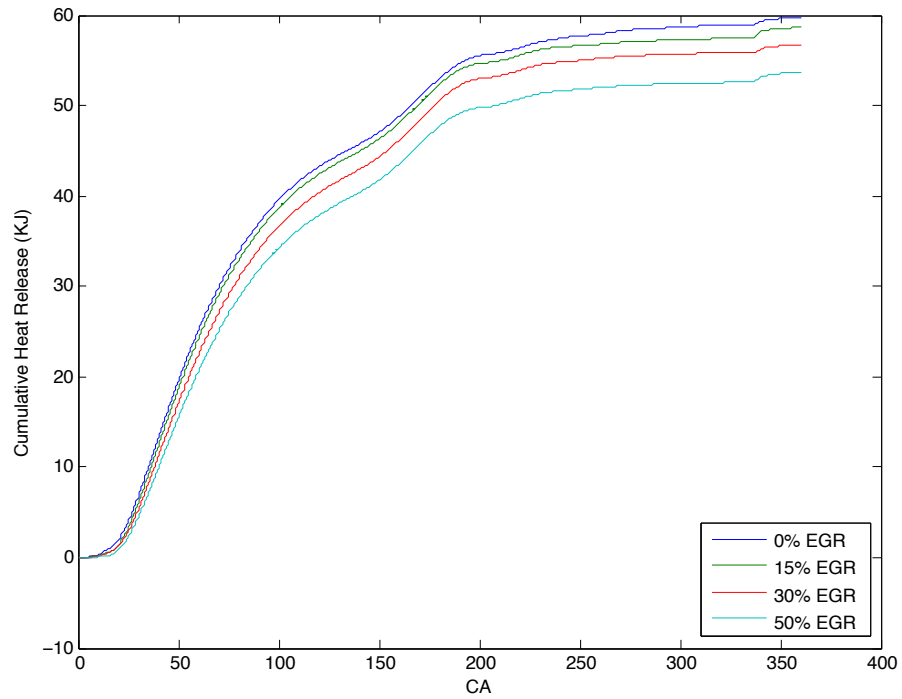
*Figure 5.18: Heat Release Rate for different EGR Rates at 30% Substitution Rate and 60% Load*



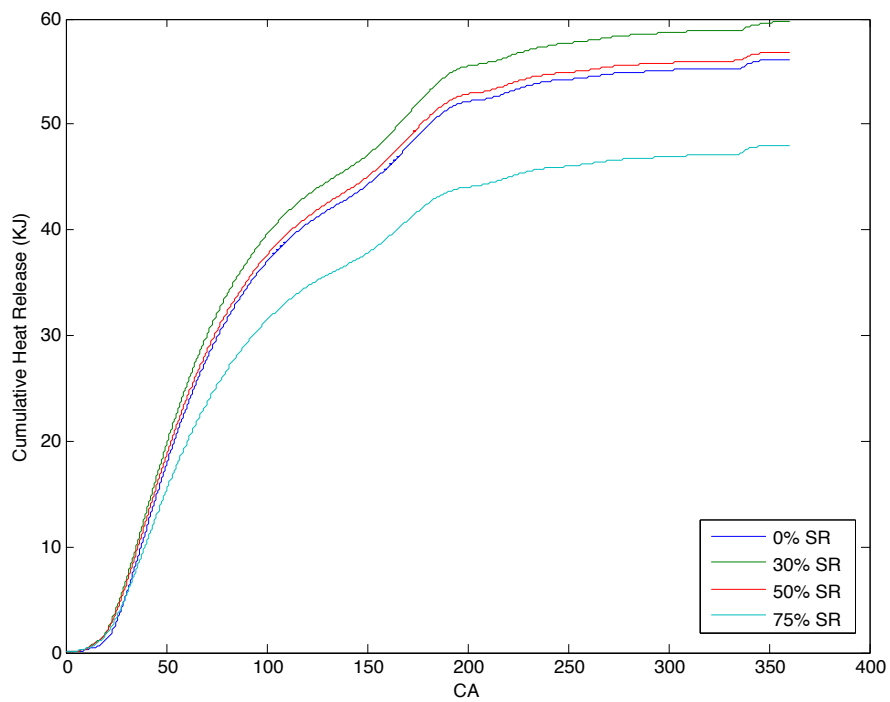
*Figure 5.19: Heat Release Rate for different Substitution Rates at 0% EGR Rate and 60% Load*



*Figure 5.20: Cumulative Heat Release for different EGR Rates at 30% Substitution Rate and 60% Load*



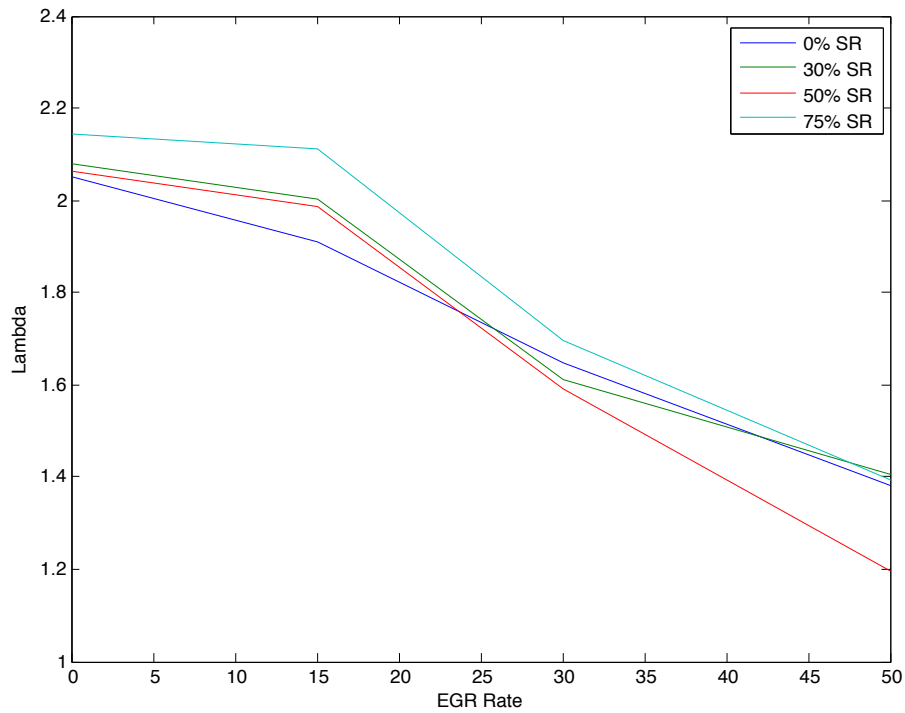
*Figure 5.21: Cumulative Heat Release for different Substitution Rates at 0% EGR Rate and 60% Load*



The figures 5.18 and 5.20 represent an undeniably decrease of the energy released for different EGR percentages during 30% of diesel substitution rate. It can be appreciated how for low EGR rates the energy degradation is almost negligible as a result of the impact promoted by the chemical and thermal effect favoring the adverse reaction brought by the weak lack of oxygen.

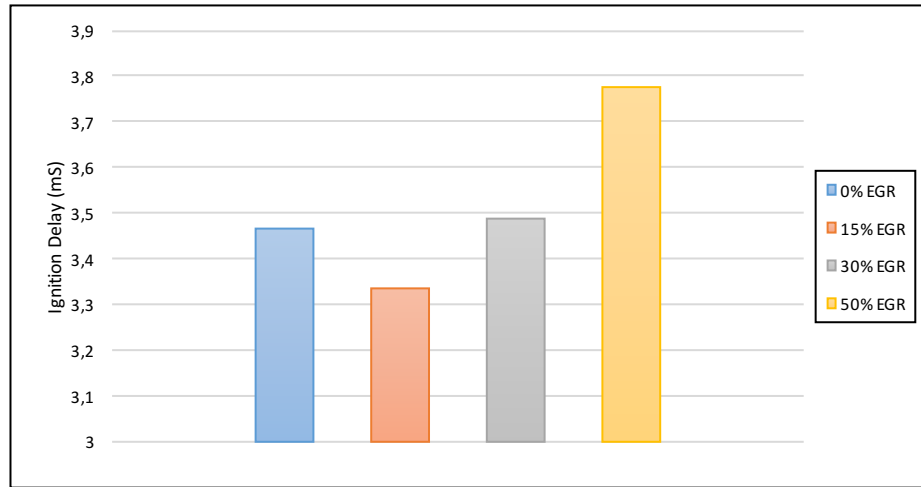
Regarding the ignition delay timing found during different EGR rates, there exists an evident similarity compared to the results obtained for lower loads, which strengthens the reliability of the impact promoted by the exhaust gas recirculation.

*Figure 5.22: Air-Fuel Equivalence Ratio for different EGR and Substitution Rates at 60% Load*

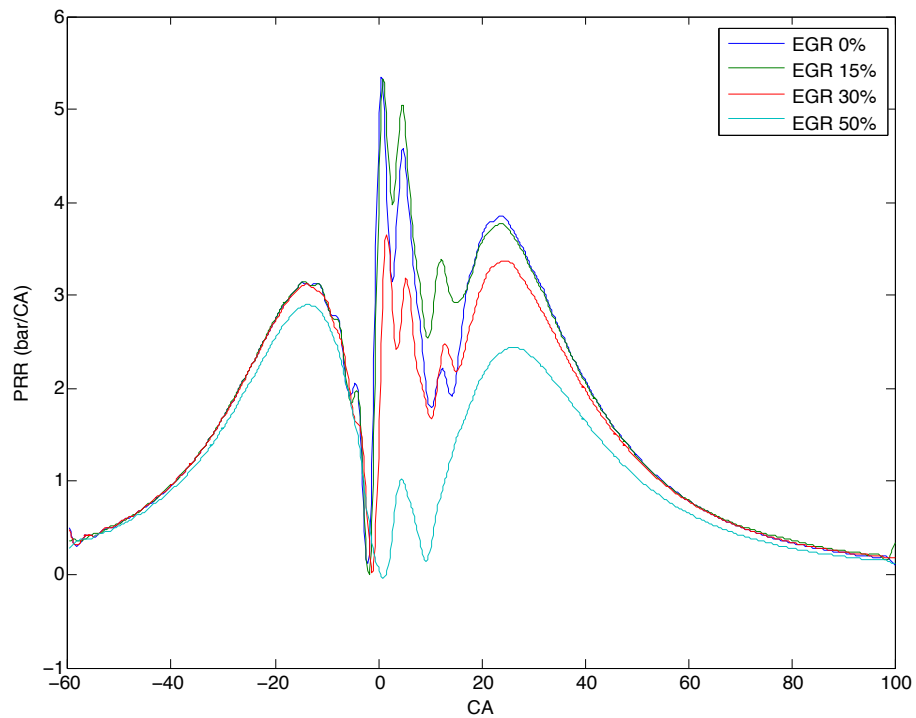


The evolution of  $\lambda$  during the operation at 60% of the maximum load is graphed in figure 5.22. The effect of the oxygen reduction contained within the intake blend straightaway undergoes a diminution of the air-fuel equivalence ratio since it is directly dependent on the pure air mass flow incoming to the cylinder. As can be appreciated, reductions up to 43% of the original value are registered for diesel substitution rates of 50%.

*Figure 5.23: Ignition Delay for different EGR Rates at 50% Substitution Rate and 60% Load*



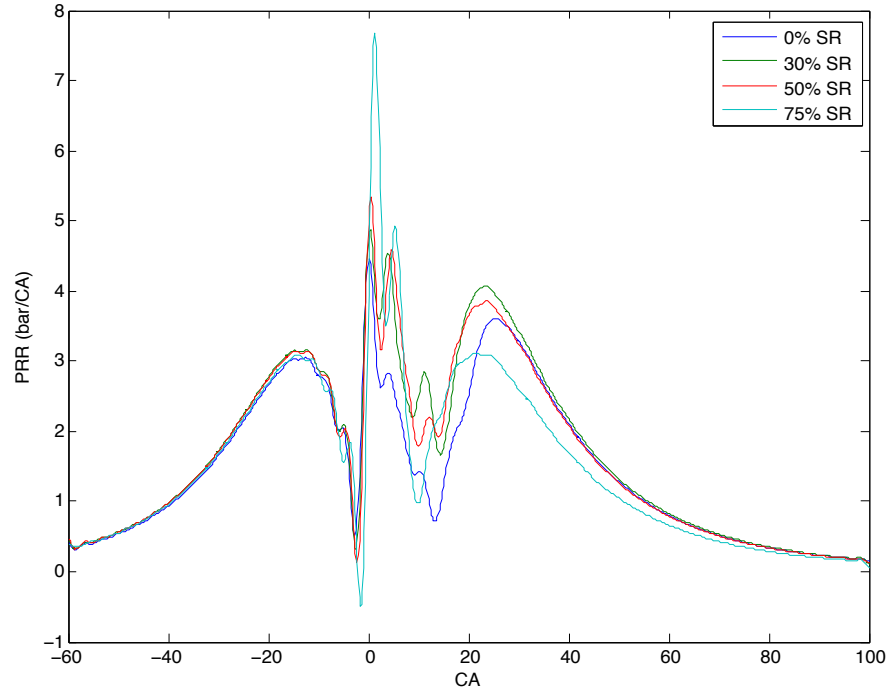
*Figure 5.24: Pressure Rise Rate for different EGR Rates at 50% Substitution Rate and 60% Load*



In the figure 5.24 it could be appreciated the attenuation of the PRR promoted by the partial recirculation of exhaust gases. This effect is widely interpreted as a reduction on the engine noise as well as the likelihood of finding knocking.

On contrary, analyzing the effect of increasing the diesel substitution rate during the dual-fuel operation, it is easily recognizable the appearance of higher PRR inasmuch the methane share grows. This effect might be explained due to the sudden auto-ignition of part of the gas-air mixture during the rapid pilot energy release. This intense burning leads to very high pressure differences reaching, in this case, values up to 8 bar/CA (even with the application of data filtering) as can be seen in the figure 5.25. However, analyzing the pressure averaged data, values over 13 bar/CA were found in the regions near to the TDC, effect that might be interpreted as the onset of knocking.

*Figure 5.25: Pressure Rise Rate for different Substitution Rates at 0% EGR Rate and 60% Load*



As previously explained, the EGR strategy is found to be as well responsible of drastic reduction of  $NO_x$  emissions. It can be appreciated how higher values of diesel substitution rate lead to lower  $NO_x$  values and correspond to higher CO and  $CH_4$  emissions as illustrated in figures 5.27, 5.28. It must be clarified the emission module cut off the reading of the actual carbon monoxide emissions for EGR rates above 50% since the maximum value the machine is capable of measuring was overtaken.

Figure 5.26:  $NO_x$  emission for different EGR and Substitution Rates

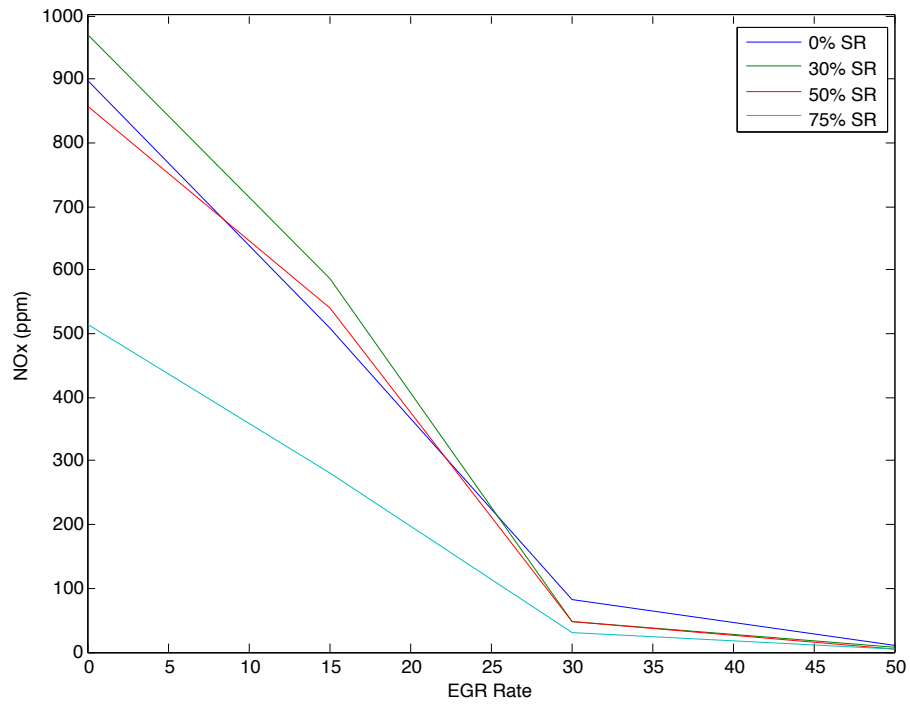
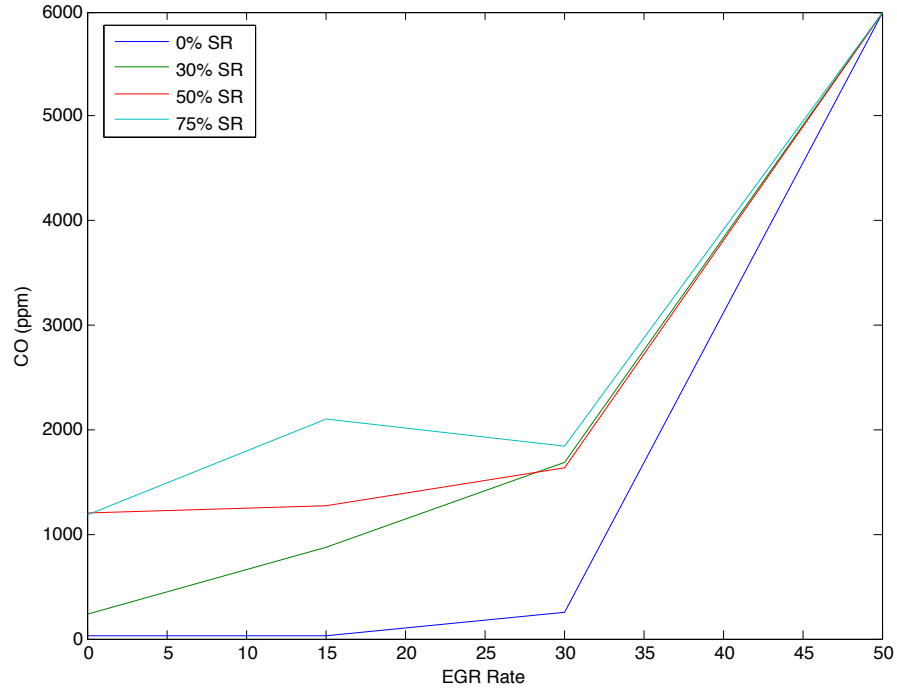


Figure 5.27:  $CO$  emission for different EGR and Substitution Rates

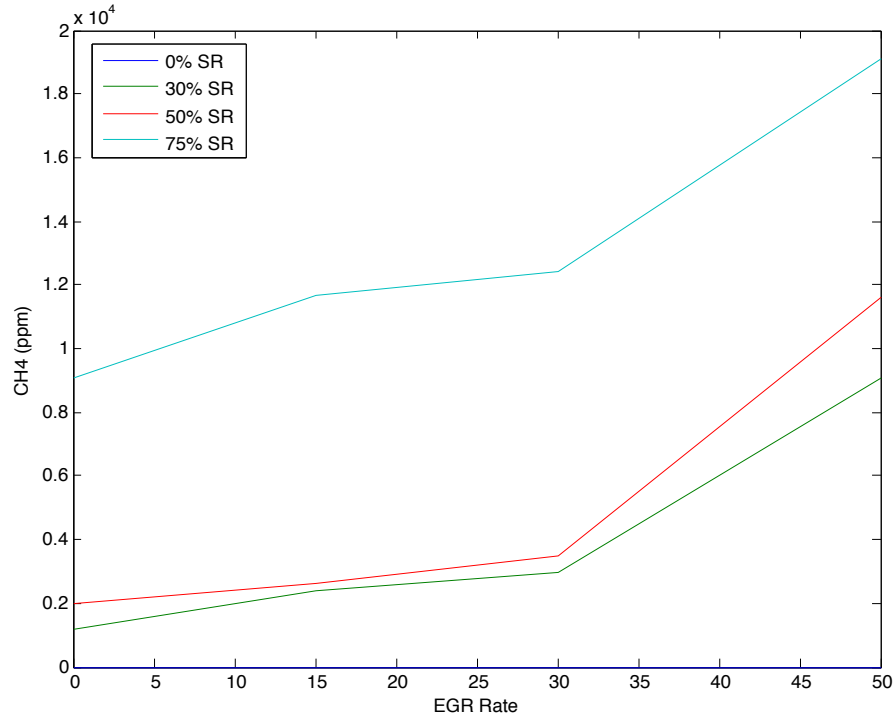


It is interesting to see the high  $CH_4$  emissions produced during 75% diesel substitution



rates even for values of 0% EGR as a result of a low combustion quality characterized by a poor flame propagation throughout the lean mixture.

*Figure 5.28: CH<sub>4</sub> emission for different EGR and Substitution Rates*



### 5.3. 70% LOAD OPERATION

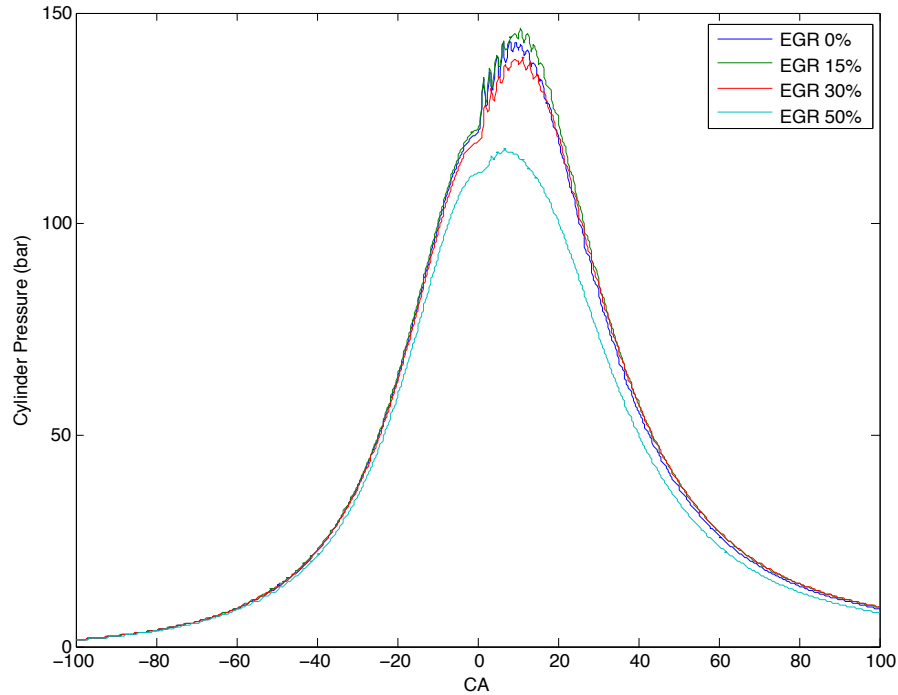
Tests made on 70% load are shown in table 5.8. In this case, as well as for 60% load, the EGR rate variances were chosen to be as appear in table 5.2.

Due to the similarity with the results obtained on 50% and 60% load, figures will be swiftly commented highlighting the most particular effects when applicable.

*Table 5.8: 70% Load Conditions*

	Charge Air Pressure (bar)	Back Pressure (bar)	Energy Content (KJ/s)	$\dot{m}_{\text{diesel}}$ (Kg/h)	$\dot{m}_{\text{methane}}$ (Kg/h)
0% Substitution Rate	2	2,5	268	6,1	0
30% Substitution Rate	2	2,5	268	4,253	1,54
50% Substitution Rate	2	2,5	268	3,225	2,5
75% Substitution Rate	2	2,5	268	1,531	3,65

*Figure 5.29: Cylinder Pressure for different EGR Rates at 75% Substitution Rate and 70% Load*

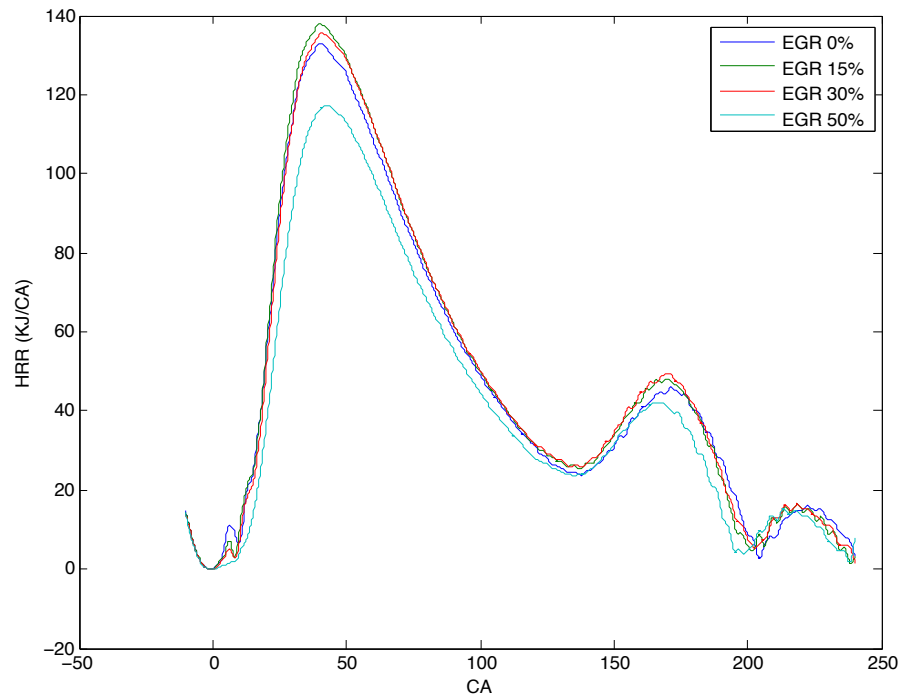


In this case, the analysis of the cylinder pressure curve represents a very similar behaviour concerning EGR rates up to 30%, showing a clear reduction when EGR rates of 50% and above are taken into consideration.

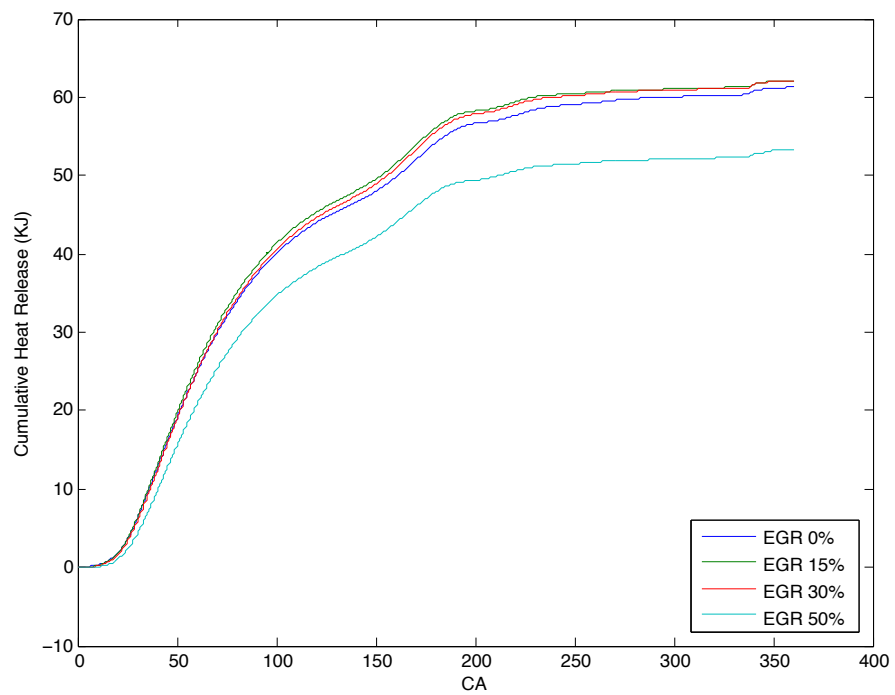
According to this result, it appears the effect of recirculating part of the exhaust gases is slightly less obtrusive than the effect experience on lower loads.

As could be assumed, this minor differences on cylinder pressures will not produce a remarkable effect when looking at the heat release curves, which are graphed in figures 5.30 and 5.31. As might be expected, deviations regarding 50% of EGR rates are registered, while no notorious discrepancies are found for the rest operating points.

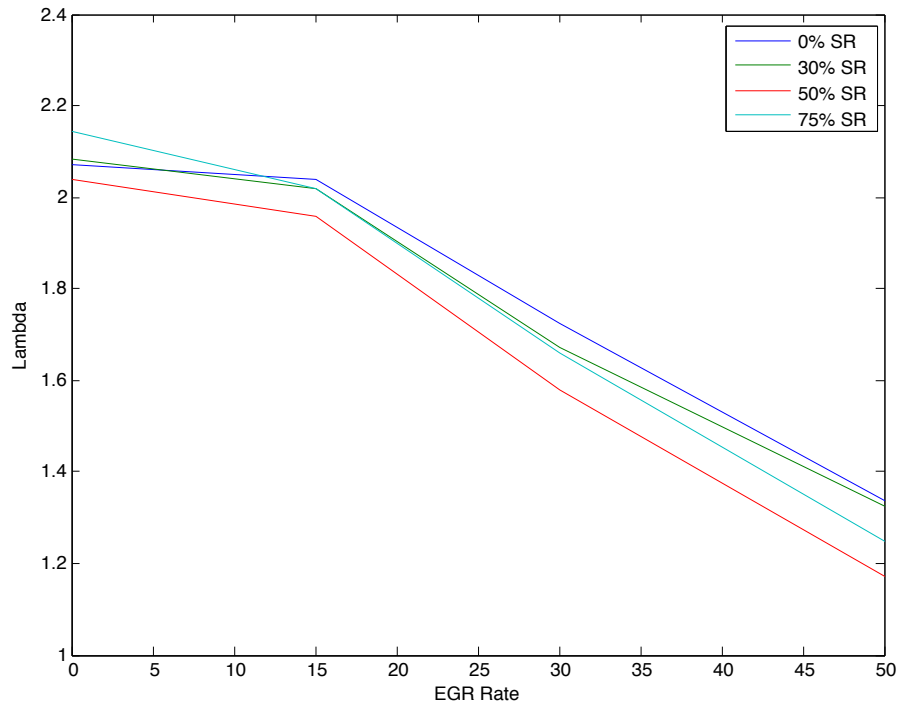
*Figure 5.30: Heat Release Rate for different EGR Rates at 75% Substitution Rate and 70% Load*



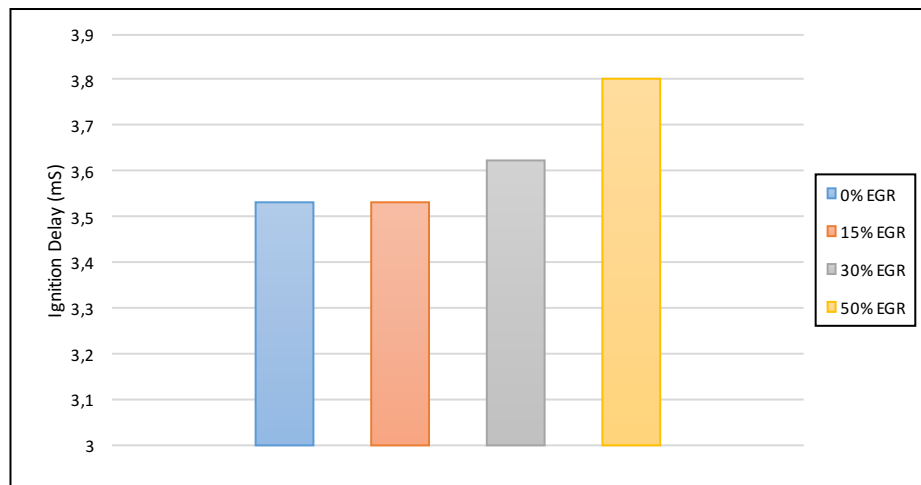
*Figure 5.31: Cumulative Heat Release for different EGR Rates at 75% Substitution Rate and 70% Load*



**Figure 5.32:** Air-Fuel Equivalence Ratio for different EGR and Substitution Rates at 70% Load



**Figure 5.33:** Ignition Delay for different EGR Rates at 75% Substitution Rate and 70% Load

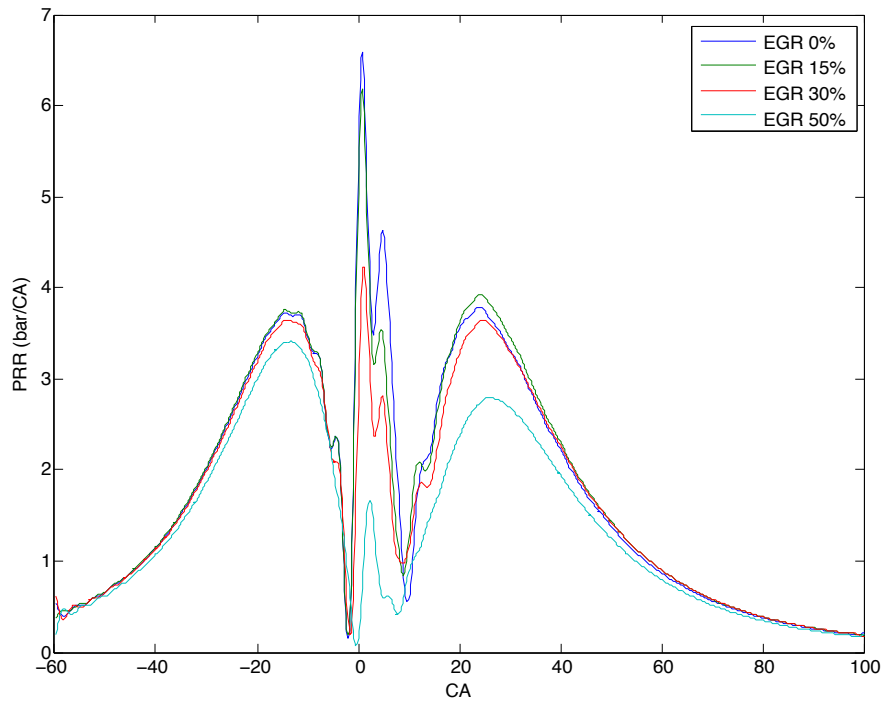


Naturally, the trends registered in the emissions (figures 5.35, 5.36 and 5.37) show certain similarity with the results achieved during the experiments at lower loads (50-60%), pointing rapid decreases on  $NO_x$  as well as exponential increases in both CO and  $CH_4$  emissions.

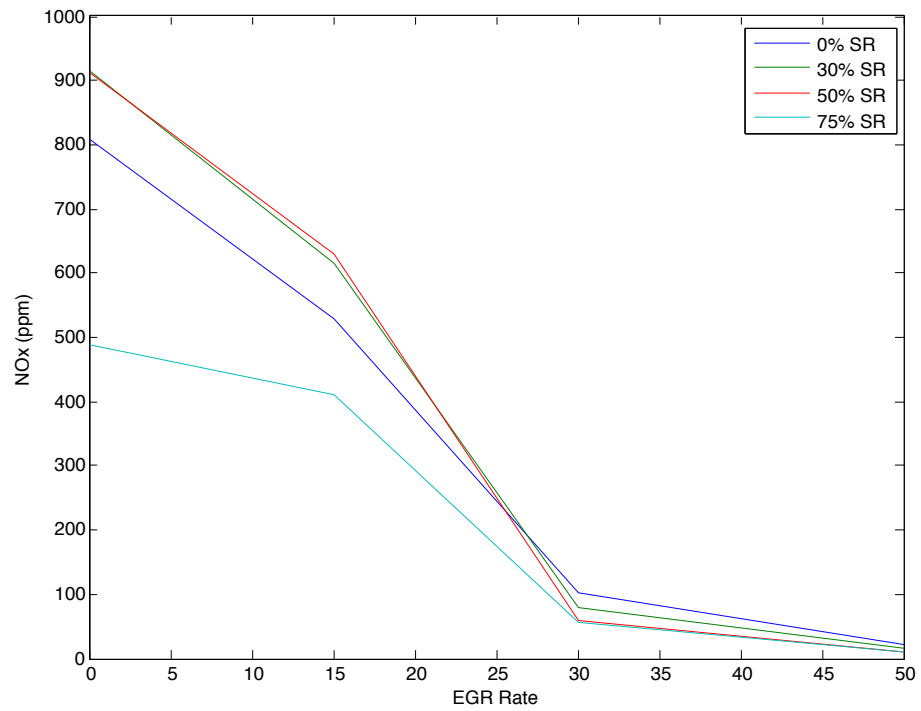
Yet EGR rates of 15 and 30% bring enormous reduction of  $NO_x$  emissions (as can be seen in figure 5.35) without compromising the thermal efficiency nor CO and  $CH_4$  emissions excessively.

Therefore, the positive effect produced by the utilization of EGR rates up to 30% regarding exhaust  $NO_x$  and energy released during operation at 70% of the maximum load must be taken into consideration when it comes to decide whether implement EGR strategies into the internal combustion engines field (and more specifically on dual-fuel mode) or not, despite the presence of drastic growths in carbon monoxide and  $CH_4$  emissions.

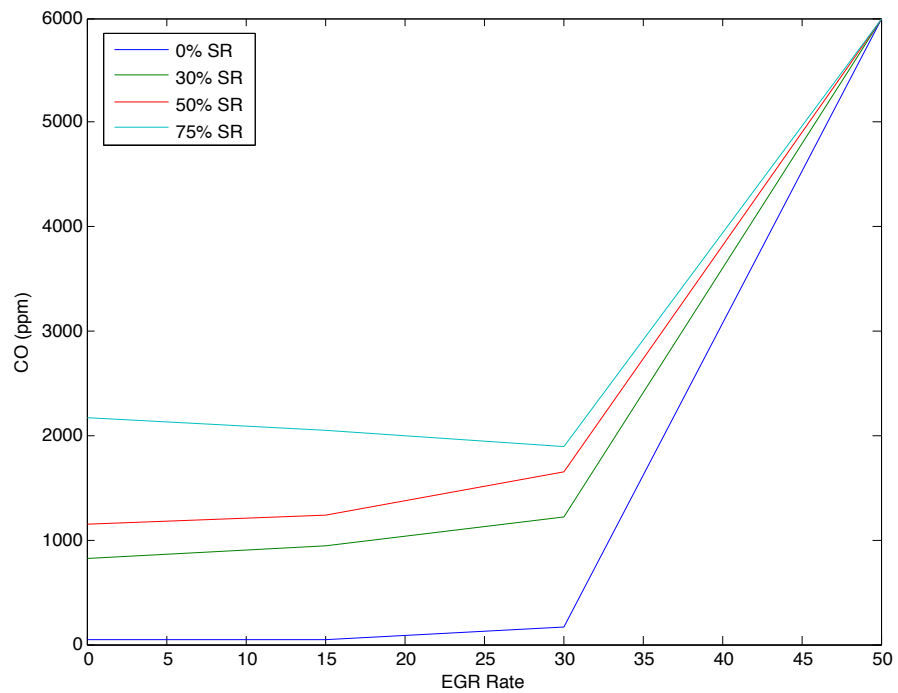
*Figure 5.34: Pressure Rise Rate for different EGR Rates at 75% Substitution Rate and 70% Load*



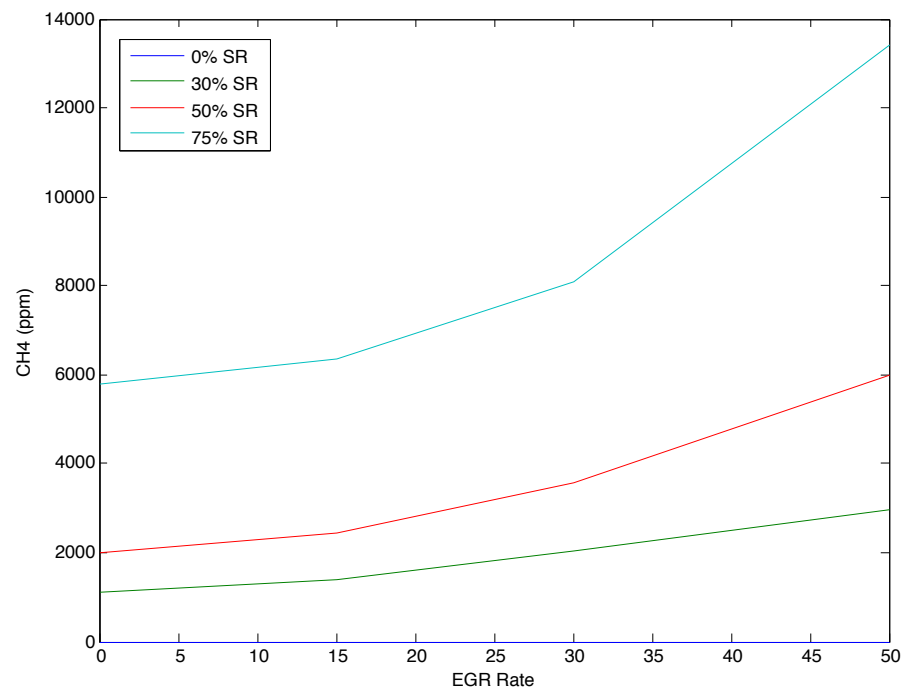
*Figure 5.35: NO<sub>x</sub> emission for different EGR and Substitution Rates*



*Figure 5.36: CO emission for different EGR and Substitution Rates*



*Figure 5.37: CH<sub>4</sub> emission for different EGR and Substitution Rates*



## 6. CONCLUSIONS

The aim of this master thesis is to give sense to the utilization of exhaust gas recirculation systems into the dual-fuel operation.

The anti-environmental effects brought by pollutant emissions along with the ongoing depletion of fossil fuels have served of motivation to humankind for the utilization of different strategies concerning the internal combustion engine field, pointing its evolution to the development of different alternatives with the objective of achieving reduction on pollutant emissions as well as producing enhancements on the engine benefits.

In this sense, utilization of EGR systems during dual-fuel operation can play an important role and must be considered when it comes to decide what type of engine strategy to use since it may lead to beneficial effects in thermal efficiency and emissions but also could produce substantial profits economically speaking.

As demonstrated along this study, exhaust gas recirculation into dual-fuel engines promote, if adequate conditions are set, gradual benefits regarding thermal efficiency as well as decreasing the exhaust emissions, being the main fact the reduction of  $NO_x$  during the operation. It is proved how low EGR rates of about 15% does not greatly compromise the released energy, but drastically reduce  $NO_x$  up to 10 times the original value. The drawback is found when high EGR percentages are in use since it encourage exponential increases on carbon monoxide and unburned hydrocarbons emissions due to the incomplete fuel oxidation produced by high dilution effects.

The effect of varying the diesel substitution rate was as well analyzed during this research. Substitution rates of 30 and 50% of the diesel share were found to bring higher cylinder pressures and heat release, which may be understood as increasing the power output during the cycle. The substitution of part of the diesel share by methane was also responsible of producing lower  $NO_x$  emissions along with higher CO and  $CH_4$  emissions (null when 100% diesel share utilized). Notwithstanding, special attention should be paid to this scheme since the increase of the methane share is closely related with the onset of knocking.

The utilization of both EGR strategies and dual-fuel operation in internal combustion engines lead to several impacts affecting to a large degree the engine operation. These effects may be summarized as follows:



- Slight enhancement of thermal efficiency associated to increases of the heat release.
- Large reductions of  $NO_x$  emissions as a trade-off of increasing the CO and UHC emissions.
- Increase on ignition delays promoted by growths of the methane share and the dilution effect found during high EGR operation.
- Lower values of rates of pressure rise as safeguard of knocking for high EGR percentages and the opposite trend while increasing the methane share.
- Appearance of misfiring at large EGR rates and knocking for high diesel substitution rates.

# Bibliography

- [1] GH Abd-Alla, HA Soliman, OA Badr, and MF Abd-Rabbo. Effects of diluent admissions and intake air temperature in exhaust gas recirculation on the emissions of an indirect injection dual fuel engine. *Energy Conversion and Management*, 42(8):1033–1045, 2001.
- [2] MM Abdelaal and AH Hegab. Combustion and emission characteristics of a natural gas-fueled diesel engine with egr. *Energy conversion and management*, 64:301–312, 2012.
- [3] European Comission. European crude oil imports and supply cost. 2015.
- [4] RJ Crookes, T Korakianitis, and AM Namasivayam. A systematic experimental assessment of the use of rapeseed methyl ester (rme) as a compression ignition engine fuel during conventional and dual-fuel operation. In *TAE 7th international colloquium on fuels, Stuttgart, 14e15 Jan*, 2009.
- [5] Ministerio de Industria Energía y turismo. Gobierno de España. Información sobre el gas natural, 2016. URL: <http://www.minetur.gob.es/energia/gas/Gas/Paginas/gasnatural.aspx>.
- [6] Michael J Economides and David A Wood. The state of natural gas. *Journal of Natural Gas Science and Engineering*, 1(1):1–13, 2009.
- [7] Eurostat. Producción e importaciones de energía, 2015. URL: [http://ec.europa.eu/eurostat/statistics-explained/index.php/Energy\\_production\\_and\\_imports/es](http://ec.europa.eu/eurostat/statistics-explained/index.php/Energy_production_and_imports/es).
- [8] Liquid Natural Gaz. Liquefied natural gas characteristics, 2015. URL: <http://www.naturalgaz.com/index.php?sect=10>.
- [9] Boyan Guo and Ali Ghalambor. *Natural gas engineering handbook*. Elsevier, 2014.
- [10] Alberto Murillo Hernández. *Exhaust Gas Recirculation Study on Dual-Fuel Methane Combustion*. 2015.
- [11] John B Heywood et al. *Internal combustion engine fundamentals*, volume 930. Mcgraw-hill New York, 1988.

- [12] Ghazi A Karim. Combustion in gas fueled compression: ignition engines of the dual fuel type. *Journal of engineering for gas turbines and power*, 125(3):827–836, 2003.
- [13] Ghazi A Karim. *Dual-fuel diesel engines*. CRC Press, 2015.
- [14] Fredrik Königsson. *On Combustion in the CNG-Diesel Dual Fuel Engine*. Number 2014:08 in TRITA-MMK. 2014. QQC 20140915.
- [15] T Korakianitis, AM Namasivayam, and RJ Crookes. Natural-gas fueled spark-ignition (si) and compression-ignition (ci) engine performance and emissions. *Progress in Energy and Combustion Science*, 37(1):89–112, 2011.
- [16] Jin Kusaka, Takashi Okamoto, Yasuhiro Daisho, Ryouji Kihara, and Takeshi Saito. Combustion and exhaust gas emission characteristics of a diesel engine dual-fueled with natural gas. *JSAE review*, 21(4):489–496, 2000.
- [17] Gordon P McTaggart-Cowan, HL Jones, Steven N Rogak, W Kendal Bushe, Philip G Hill, and SR Munshi. Direct-injected hydrogen-methane mixtures in a heavy-duty compression ignition engine. Technical report, SAE Technical Paper, 2006.
- [18] Santiago A Molina. *Influencia de los parámetros de inyección y la recirculación de gases de escape sobre el proceso de combustión en un motor diesel*. Reverté, 2005.
- [19] Peter L Mtui and Philip G Hill. Ignition delay and combustion duration with natural gas fueling of diesel engines. Technical report, SAE Technical Paper, 1996.
- [20] AM Namasivayam, RJ Crookes, T Korakianitis, and J Olsen. Assessment of combustion in natural gas dual-fuelled compression ignition engines with dimethyl ether and rapeseed methyl ester pilot ignition. *International Journal of Engine Research*, 10(3):165–174, 2009.
- [21] George A Olah, Alain Goeppert, and GK Surya Prakash. *Beyond oil and gas: the methanol economy*. John Wiley & Sons, 2011.
- [22] RG Papagiannakis and DT Hountalas. Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas. *Energy conversion and management*, 45(18):2971–2987, 2004.
- [23] V Pirouzpanah, R Khoshbakhti Saray, A Sohrabi, and A Niaei. Comparison of thermal and radical effects of egr gases on combustion process in dual fuel engines at part loads. *Energy conversion and management*, 48(7):1909–1918, 2007.
- [24] BB Sahoo, N Sahoo, and UK Saha. Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines—a critical review. *Renewable and Sustainable Energy Reviews*, 13(6):1151–1184, 2009.

- [25] Susan L Sakmar. *Energy for the 21st century: opportunities and challenges for liquefied natural gas (LNG)*. Edward Elgar Publishing, 2013.
- [26] Mohamed YE Selim. Effect of exhaust gas recirculation on some combustion characteristics of dual fuel engine. *Energy Conversion and Management*, 44(5):707–721, 2003.
- [27] KK Srinivasan, SR Krishnan, and KC Midkiff. Improving low load combustion, stability, and emissions in pilot-ignited natural gas engines. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 220(2):229–239, 2006.
- [28] The Engineering Tool. Fuels - higher calorific values, 2016. URL: [http://www.engineeringtoolbox.com/fuels-higher-calorific-values-d\\_169.html](http://www.engineeringtoolbox.com/fuels-higher-calorific-values-d_169.html).
- [29] Tino Tuominen. *Potential of Methanol in Dual Fuel Combustion Engines*. 2016.
- [30] Lijiang Wei and Peng Geng. A review on natural gas/diesel dual fuel combustion, emissions and performance. *Fuel Processing Technology*, 142:264–278, 2016.